THE PRINCIPLES OF HEATING

WILLIAM G. SNOW

PRINCIPLES OF HEATING

A practical and comprehensive treatise
on Applied Theory in Heating.

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PRINCIPLES OF HEATING.

PREFACE.

While the title of this book may not be sufficiently comprehensive, it perhaps expresses, as nearly as may be done in a few words, the contents of the following pages. These are largely made up of a collection of articles by the author, which have appeared from time to time during the past few years in the *Metal Worker, Plumber and Steam Fitter.*

These contributions have been supplemented by reprints of articles relating to heating prepared by other writers.

Included in this work are the results of tests made by the author on heating apparatus and systems, together with numerous original tables and charts which he has found to be of practical use in the solution of heating problems.

Considerable space is devoted to a collection of articles on Vacuum and Vapor systems of heating, in view of the amount of interest manifested of late in this class of apparatus.

Special stress has been laid on the application of the heat unit to the solving of heating problems.

It is hoped that by the aid of the complete table of contents and the index persons interested in the subject treated will find the data contained in the following pages convenient for reference.

BOSTON, 1907. WILLIAM G. SNOW.

The work has been revised, rearranged, and added to in 1912.

W. G. S.

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CHAPTER I.

HEATING POWER OF FUELS, BOILERS AND COMBINATION HEATERS.

THE HEAT UNIT.

What the pound is to the grocer, and the 2-foot rule is to the carpenter, the heat unit should be to those engaged in heating and ventilating work. It is their unit of measurement and is the common sense basis of all heating calculations. Briefly stated, a heat unit is the amount of heat required to raise the temperature of 1 pound of water 1 degree F.

To make practical use of the heat unit one must become familiar with the heating power of fuels, the loss of heat through walls

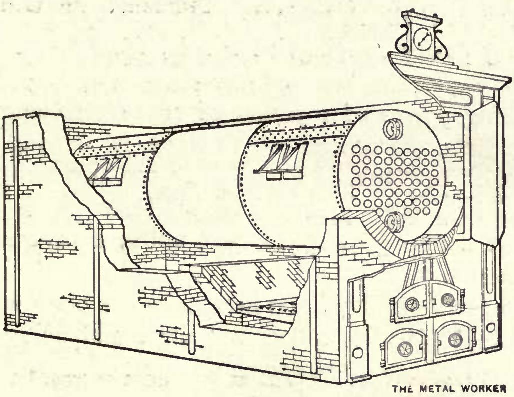

Fig. 1.—Horizontal Tubular Boiler.

and glass, the heat emitted by radiators and many other facts bearing on the subject. It is hoped this information will prove useful to those who wish to know the " whys and wherefores " of heating calculations and are not content to blindly follow " thumb rules," which may be good enough for small work, but for large undertakings are apt to give very unsatisfactory results and bring a serious loss to the contractor. A good grasp of the " heat unit basis " gives one confidence to attack and the ability to solve almost any heating problem that may arise.

THE HEATING POWER OF FUELS, ETC.

Anthracite coal has a theoretical heating power of about 14,200 heat units per pound of combustible. With 10 per cent. ash and noncombustible matter, 1 pound has a heating power of about 13,000 heat units. The smaller the coal the greater the percentage of ash, 16 per cent. or more being not uncommon with the smaller sizes.

Coke, like anthracite coal, consists almost entirely of carbon and has about the same heating power.

Good bituminous coal has a heating power of about 13,000 to 14,000 heat units per pound of combustible.

About 2½ pounds of dry wood have the same heating power as a pound of coal.

Taking a fair average, 25,000 cubic feet of natural gas, or 40,000 cubic feet of illuminating gas, are equivalent in heating power to a ton of coal. A cubic foot of ordinary illuminating gas has a heating power ranging, as a rule, from 600 to 700 heat units.

The heating power of 1 pound of crude petroleum is about 21,000 heat units, the refined oil, or kerosene, having a heating power, in round numbers, of 27,000 to 28,000 heat units.

Electrical heat units are: 1 kilowatt hour equals 3412 heat units; 1 watt hour equals 3.412 heat units; 1 heat unit equals 0.293 watt hours.

A person gives off about 400 heat units per hour, an ordinary gas burner approximately 4,000 heat units and an incandescent electric light of 16 candle-power about 190 heat units.

EFFICIENCY OF BOILERS AND COAL CONSUMPTION.

To determine the probable coal consumption in a heating boiler one must assume a certain efficiency. It is of interest in this connection to discuss briefly the efficiency and coal consumption of high pressure boilers of the types shown in Figs. 1 and 2, and to show the application of the heat unit in solving problems of this nature. A boiler horse-power is equivalent to 33,305 heat units per hour; hence 3 pounds of combustible per horse-power is equivalent to 11,102 heat units out of a possible 14,000 in round numbers, representing an efficiency of about 80 per cent. With 4 pounds of combustible per horse-power these figures

would be reduced to 8,326 and 60 per cent. respectively, the latter conforming more nearly to ordinary working conditions than does 80 per cent. Suppose a boiler evaporates 9 pounds of water per pound of coal containing about 16 per cent. ash; then 1 pound of coal will contain only about $^5/_6$ pound of combustible, or the evaporation will be equivalent to about 11 pounds of water per pound of combustible. To evaporate 1 pound of water at a temperature of 212 degrees F. into steam at the same temperature requires about 964 heat units; hence the evaporation of 11 pounds

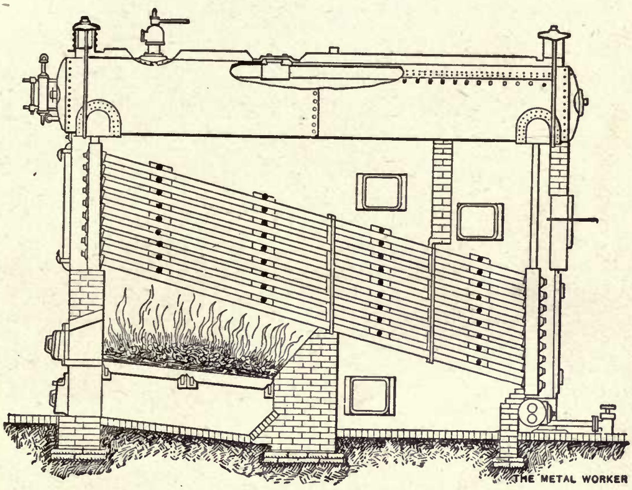

Fig. 2.—Water Tube Boiler.

of water by 1 pound of combustible is equal to 10,604 heat units per pound, or approximately 76 per cent. of the theoretical amount of heat in the coal.

Such an efficiency may be obtained under well managed high pressure boilers, but smaller cast iron heating boilers, illustrated in Figs. 3 and 4, will with the less skillful attendance given them have hardly more than 60 per cent. efficiency. In other words, we would not be likely to transfer from the fire to the water in the heater more than 8,000 to 9,000 heat units per pound of coal.

The distinction between coal and combustible must be kept in mind, the latter being only the burnable portion of the fuel.

COMPUTING GRATE SURFACE ON A HEAT UNIT BASIS.

A knowledge of the heat utilized per pound of coal burned a the total loss of heat from a building, the latter to be comput as described later, gives a convenient basis for determining t size of the heater, irrespective of the total radiating surface, which the size is commonly based. If each pound of coal burn gives up to the water in the heater 8,000 heat units; dividing t total heat loss per hour from the building by 8,000 gives t weight of coal that must be burned. The grate surface is th

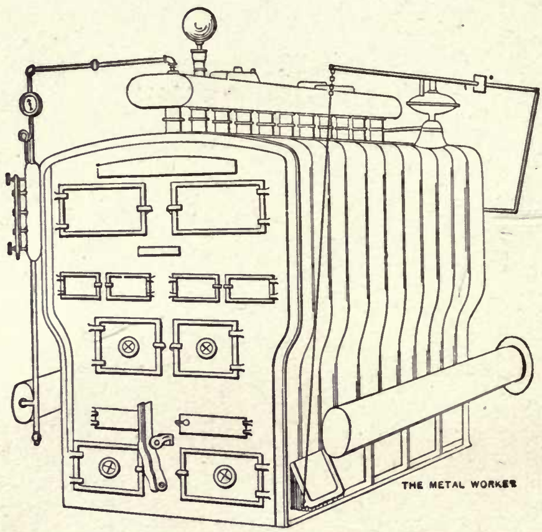

Fig. 3.—Sectional Cast Iron Boiler with Vertical Sections.

determined by dividing the weight just computed by 3 to 4 fo small boilers, 4 to 5 for those of medium size, and by 5 to 7 fo large sized boilers. These figures represent permissible rates c combustion, expressed in pounds of coal burned per square fo per hour in house heaters.

HEATING SURFACE IN BOILERS AND FURNACES.

The proper grate surface is only one element to be determine It is equally important to see that the heater selected has th proper amount of heating surface well located. As to the amour of heat absorbed per square foot of heating surface, the sma

boilers mentioned commonly have only 10 to 15 square feet of heating surface per square foot of grate, the medium sizes 16 to 20, and the larger ones 20 to 25. These proportions, with the rates of combustion stated, give from 2,000 to 2,200 heat units absorbed per hour per square foot of heating surface.

Hot air furnaces commonly have 15 to 20 square feet of heating surface to each square foot of grate. Taking the average, 17½, and a 5-pound rate of combustion, the heat emitted per square foot of heating surface would be $\frac{5 \times 8{,}000}{17.5} = 2{,}400$. This figure is, of course, only approximate, the kind and location

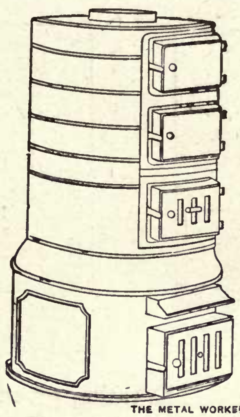

Fig. 4.—Sectional Cast Iron Boiler with Horizontal Sections.

of the heating surface making some portions more effective and others less so than the average. The heat given off varies also with the rate of combustion, but not at all in proportion to it.

HOT WATER COMBINATION HEATERS.

At best it is difficult, in a combination system of heating, to secure a proper balance between the hot water and hot air. Much depends on the proper rating of the coil or special casting used for heating the water. A number of tests made by the writer on various types of these heaters have established ratings which may safely be used in proportioning systems of this kind. In making the tests, radiators were arranged so that the total amount of surface connected with the heater could be nicely regulated to de-

termine the total radiating surface that could be maintained at an average temperature of about 160 degrees for hours at a time with an even fire and an ordinary rate of combustion.

DOME HEATER.

A dome shaped cast iron section, of the general type illustrated in Fig. 5, proved capable of maintaining an average temperature in the flow pipe of about 160 degrees when supplying approximately 15 square feet of radiating surface to each square foot of heating surface. A great increase in capacity was noted when the fire was bright on top, the heater then being subjected to the direct rays from the burning coal. At other times it was merely

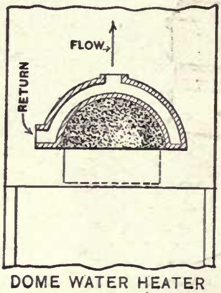

Fig. 5.—Type A.

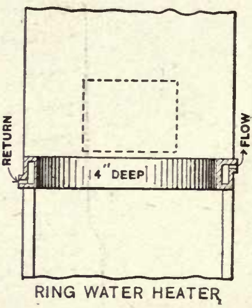

Fig. 6.—Type B.

surrounded by hot gases. The rating given is that under average conditions during an eight-hour run.

RING HEATERS.

A fire pot having a cored space 4½ inches high by about 1 inch wide extending around the entire circumference, as shown in Fig. 6, was next tested. Three tests, each of about eight hours' duration, showed this type of combination heater, having a total of 5 square feet of heating surface, to be capable of heating to an average temperature of 170 degrees, the water in the flow pipe connected with 250 square feet of direct radiation. This is equivalent to a capacity of 50 square feet of direct radiation to every square foot of heating surface in contact with the fire. A combination of the ring and the dome shown in Fig. 7, has the heating capacity stated in Table I.

Another combination heater of a similar type, shown in Fig. 8,

was tested, the cored portion of the fire pot being 8 inches high, or about two-thirds the depth of the fire. Three eight-hour tests proved these heaters capable of heating the water in the flow pipe to about 160 degrees when connected with approximately 300 square feet of radiation. This heater had nearly twice the surface of the one previously described, yet the radiating surface carried was only about 20 per cent. more and was not so hot. Only about 30 square feet of radiating surface was supplied per square foot of heating surface exposed to the fire. The rapid falling off in efficiency was due to the chilling effect on the fire of so large a body of water, necessitating more frequent attention than with the

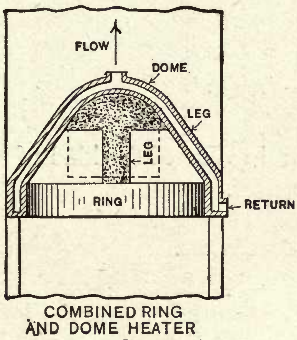

Fig. 7.—Type C.—A Combination of A and B.

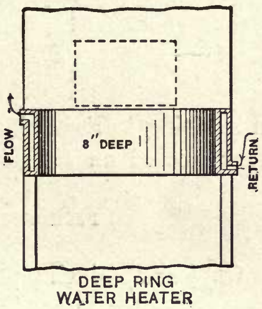

Fig. 8.—Deep Form of Type B.

other combination heaters tested. The average rate of combustion during the tests was about 3½ pounds of hard coal per square foot of grate surface per hour.

In each of the three series of tests the drop in temperature between the flow and return pipes was, on an average, about 20 degrees and remained nearly uniform throughout.

VERTICAL SLAB SECTIONS.

Some makers who use vertical hollow cast iron slabs in connection with brick lined furnaces rate them as high as 75 square feet of radiating surface per square foot of heating surface. This rating is 50 per cent. greater than the highest one stated above. With a brisk fire there is no question that a square foot

of heating surface in direct contact with the fire will carry at least 75 square feet of heating surface, but it seems hardly wise to

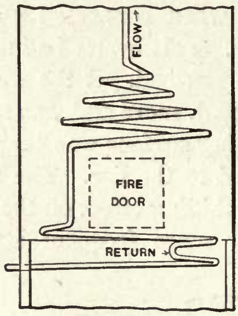

Fig. 9.—Type F.

reckon on its doing so right along, in view of the kind of attention commonly bestowed on furnace fires.

PIPE COIL HEATERS.

Coils of 1¼ or 1½-inch pipes, as shown in Fig. 9, make an excellent form of heater to combine with furnaces, especially if arranged so that the lower portion may be either above the fire or buried in it, according to the hight at which the fire is carried,

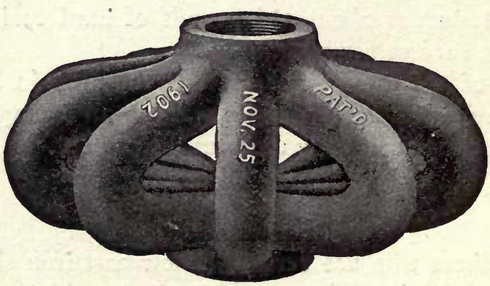

Fig. 10.—Auxiliary Heater for Combination Heating.

thus giving a ready means of regulating the temperature of the water; since the coil, when in contact with the fire, is about twice

as effective as when the fire is kept several inches below it. Pipe coils, when suspended above the fire, may be rated to carry from 20 to 25 square feet of radiating surface per square foot of heating surface, and say, 30 to 40 square feet when arranged as described, the lower strand of the coil to be in contact with the fire. Single coils placed in the fire will carry at least 50 square feet of surface per square foot of coil.

Work installed on the basis of the figures above given has

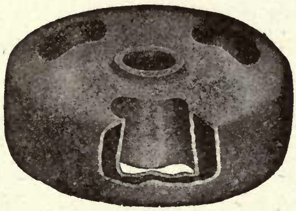

Fig. 11.—Disk Heater for Combination Heating.

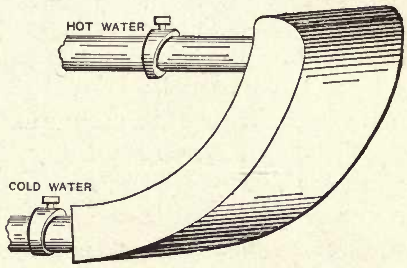

Fig. 12.—Overhanging Type of Auxiliary Heater.

proved satisfactory under the practical working conditions found in dwellings:

TABLE I.

SUMMARY, GIVING RATINGS FOR DIFFERENT CLASSES OF COMBINATION HEATERS.

Description.	Rating expressed in the number of square feet of direct radiating surface, which may be kept at a temperature of 160 degrees per square foot of heating surface in the combination heater.
A.—Cast iron sections suspended above the fire	15 to 20
B.—*Cast iron sections in contact with the fire	40 to 60
C.—A and B combined	25 to 35
D.—Pipe coil suspended above the fire	20 to 25
E.—Pipe coil buried in the fire	50 to 60
F.—D and E combined	30 to 40

* Capacity decreases as the depth of the surface in contact with the fire is increased, since the deeper the section the greater the chilling effect of the water on the fire and the harder to keep up the latter.

TYPES OF COMBINATION HEATERS.

Several common types of combination heaters on the market are shown in Figs. 10, 11 and 12.

CHAPTER II.

GAS, OIL, AND ELECTRICITY vs. COAL, AND THE CAPACITY AND FUEL CONSUMPTION OF HOUSE HEATING BOILERS.

The question sometimes comes up whether to use gas or oil, instead of coal, for heating purposes. On a heat unit basis, we may not expect to utilize more than 8,000 to 9,000 units from each pound of coal burned. Comparing this with gas having a heating power of about 700 heat units per cubic foot, and assuming that 75 per cent. of the heat is transferred to the water in the heater, we have 525 heat units utilized per cubic foot of gas burned. From a ton of coal there would be utilized 2,000 (lbs.) × 8,500 (heat units, as a maximum) = 17,000,000 heat units. This amount divided by 525 gives 32,400 cubic feet, or the equivalent amount of gas in heating effect. This volume of gas, at $1 per 1,000, would cost more than five times as much as a ton of coal at $6 per ton having the same heating power. Of course, the great advantages possessed by gas over coal are the absence of dirt and the ability to instantly turn on or shut off the heat.

The following statement by B. T. Galloway,* who made a number of experiments on oil and gas heating, is of interest:

"Oil (and by this material we mean the refined product, kerosene) may be dismissed with a few words, as, after many trials with numerous devices, it is found to be impracticable as a means of heating water or generating steam. In all of our experiments oil and gas were used to heat water circulating either in pipes or ordinary radiators. Taking an ordinary heating plant, say with a radiating capacity of 500 to 1,000 square feet, oil, when burned in the boiler with any of the so-called hydrocarbon burners, would be beyond the means of the ordinary house owner. The cost of heating 500 square feet of radiation, using kerosene oil and the best devices we have been able to secure or make, would be about three times as great for oil as compared with anthracite coal, provided coal was selling at $6 per ton delivered in the cellar, and oil at 10 cents per gallon delivered in the same way. Then the

* See "Heating Experiments with Oils and Manufactured Gas," by B. T. Galloway, in *The Metal Worker, Plumber and Steam Fitter,* October 17, 1903.

labor of handling oil, watching the burners and keeping the apparatus in order is fully as great as that connected with putting on coal and taking out ashes. Furthermore, we have never seen an oil device that could be entirely trusted, as experience with them shows that, when least expecting it, they go wrong, and fire and explosion follow unless great care is observed. The utilization of oil, therefore, as described, is hardly to be recommended.

"There is one method of utilizing oil, however, which is worthy of further trial and consideration—viz., that of adopting as a burner the ordinary blue flame oil stove, of which there are

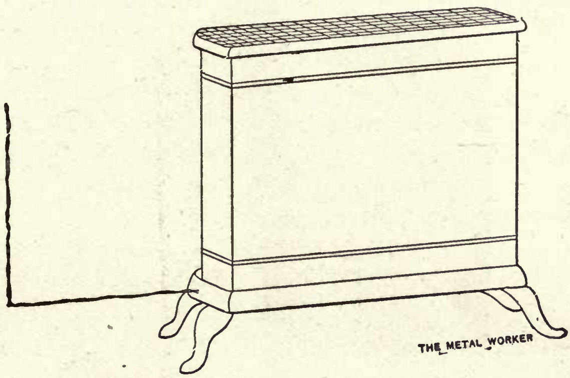

Fig. 13.—Type of Electric Radiator.

several kinds on the market. The burners for these stoves can be bought separately. They have a gravity feed, and will run indefinitely with little care and attention.

"It was found that boilers made for coal with their arrangements for cinders, drafts, etc., were poorly adapted for the use of a fuel as costly as gas. Only a small portion of the efficient heat units in the gas could be utilized, the rest going up the chimney or being lost in overcoming the resistance offered by the iron and in other ways. With specially constructed boilers, and by such we mean those where the flame of the burning gas can be brought into direct contact with a large surface of some metal like copper, much more effective results can be obtained than where ordinary boilers made for coal are used. Types of such boilers are to be found in those used for automobiles containing either a large number of small copper tubes or consisting of series upon series of

copper coils through which the circulating water passes. Even with such devices, however, it has been found impracticable to sufficiently heat the water from a central plant, except at a cost considerably more than that of coal at ordinary prices. In actual practice the cost of the gas would be about double that of coal, the price of the latter being estimated at $6 per ton, and the former at $1 per 1,000 cubic feet, 22 candle-power. Of course, in this case there is no coalman to bother with, no ashes to take out and no trouble in regulating the apparatus with the proper de-

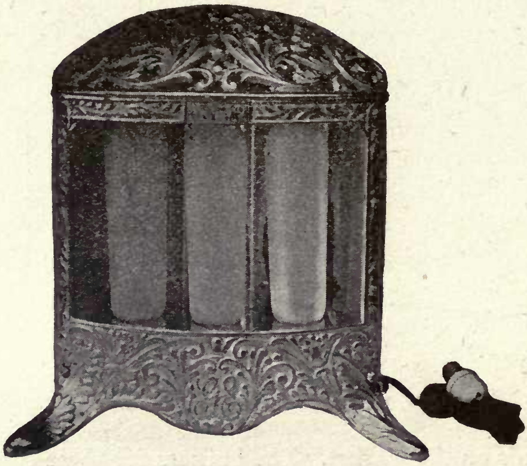

Fig. 14.—Luminous Electric Radiator.

vices at hand. Theoretically, and practically too for that matter, there is no reason why a householder could not light his burner in the autumn and the apparatus would do the rest, until it was time to turn the gas off in the spring. By means of properly adjusted regulators, the gas would be fed to the burner in sufficient amounts to maintain a uniform temperature in the room above. With gas at present prices this method of heating would be practically prohibitive for many, notwithstanding its advantages."

Extracts from an article by Donald McDonald on "Domestic Heating by Gas"* seem worth repeating here:

* See "Domestic Heating by Gas," Donald McDonald, in *The Metal Worker, Plumber and Steam Fitter*, October 24, 1903.

"Where the gas is the only source of heat and the room is occupied as a bed chamber it is much better, although somewhat more expensive, to use a closed heater provided with a good flue. Such a heater must, however, meet many very rigid conditions; otherwise the flue connection will be worse than useless. First of all, the flue must be so open and must run so high that a down draft through it will be an impossibility. A few seconds of down draft, carrying with it a load of carbonic acid and nitrogen, will put out the fire, and the flue becoming cold, the down draft will continue and the apartment become full of gas. No flue at all is

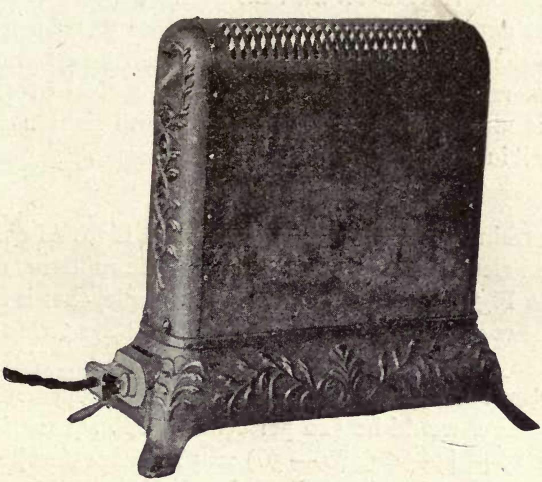

Fig. 15.—Non-Luminous Electric Radiator.

much better than this. The stove must also be so constructed that no more air is drawn through it than is necessary to burn the gas, otherwise there will be a great waste of heat up the chimney.

"The amount of air required to burn the gas, if it is cooled to 300 degrees before it reaches the chimney, will only carry away with it about 5 per cent. of the heat. Closed stoves, however, as generally constructed, send up the chimney anywhere from 20 to 80 per cent. of the heat produced by the gas. Any device which sends a part of the products of combustion up the chimney and the rest of it into the room is simply folly. The part which reaches the chimney is no better and no worse than the part which is

put into the room, and unless care is taken to send all the products of combustion up the chimney it is much more sensible not to send any of them.

"I have seen and heard many learned discussions as to the question of whether a luminous flame or a blue flame produces the most heat. Nearly all salesmen and dealers of gas stoves will insist that the particular burner which they are advocating produces a great deal more heat than any other burner. Of course, any chemist or any engineer knows that if the combustion is complete and all the products of combustion escape into the room to be heated, the room receives all the heat due to the combustion of the fuel, and no amount of ingenuity can increase this 1 per cent. If the combustion is not complete the odor will be so vile that no one will tolerate it. In other words, in this class of stoves the efficiency is almost always 100 per cent., and need not be considered at all in selecting them."

ELECTRIC HEATING.

To determine, on the heat unit basis, what it would cost to heat a room with electricity by means of an electric radiator, or heater, as shown in Fig. 13, let us suppose, for example, that it is desired to know the cost of heating a corner room, 14 x 14 x 10 feet, ten hours per day under average weather conditions. With, say, 20 per cent. glass surface, the equivalent glass surface, corresponding to the exposure, would be (20 per cent. of 280 square feet = 56 square feet) + [$\frac{1}{4} \times (280 - 56) = 56$ square feet] = a total of 112 square feet of glass; wall surface being rated as one-fourth as much glass surface. One hundred and twelve square feet of glass $\times$ 85 heat units per square foot an hour for 70 degrees difference in temperature $\times$ 1.25 (the factor for northwest exposure) = approximately 11,900 heat units per hour.*

A certain allowance must be added for quickly warming the contents of the room, apart from the transmission loss above computed. To do this it is convenient to add to the computed loss of heat through walls and windows a number of heat units equal to at least one-third the cubic contents; in this case 1-3 $\times$ 1,960 = 653 heat units. This combined with the 11,900 heat units previously computed, gives a total of 12,553 heat units per hour.

* See page 43 and following for a fuller discussion of computation of heat losses.

Electric current, when metered, is charged for on the basis of watt hours, a heat unit being equivalent to 0.293 watt hour. Therefore, 12,553 heat units would be equivalent to 3,680 watt hours; or, to heat the room ten hours in zero weather by electricity would require 36,800 watt hours.

The average amount, during the heating season, would probably not exceed, for a ten-hour day, $\frac{30}{70} \times 36{,}800 = 15{,}800$ watt hours, approximately. Ten cents per 1,000 watt hours is a not uncommon rate for such service; and at this price the cost to heat the room ten hours per day in average weather would be $1.58, a prohibitive cost.

With coal, such a room, with a 50 square foot steam radiator, would, in zero weather, allowing 250 heat units per square foot of radiating surface per hour and 8,000 heat units per pound of coal, take only 50 (square feet) $\times$ 250 (heat units) $\times$ 10 (hours) $\div$ 8,000 $=$ 15.6 pounds of coal, costing, say, 5 cents.

Electric heating is bound to be expensive in comparison with steam, if the exhaust from the power plant goes to waste, since about 90 per cent. of the heat of the steam passes away with the exhaust from the engines. With, say, 75 per cent. boiler efficiency, 10 per cent. engine efficiency on a heat unit basis and 85 per cent. on a mechanical basis (that is, allowing 15 per cent. for friction) and 90 per cent. dynamo efficiency and 95 per cent. line efficiency, we have for the combined efficiency of boiler, engine, dynamo and wires: $0.75 \times 0.10 \times 0.85 \times 0.90 \times 0.95 = 5.45$ per cent. The efficiency of a direct steam heating system would probably be as high as 55 to 60 per cent., or, say, 10 times that of the electric heating system.

THE CAPACITY AND FUEL CONSUMPTION OF HOUSE HEATING BOILERS.

Manufacturers' boiler ratings vary so widely that it is worth while for contractors to compute the capacities themselves and not trust implicitly the figures given in the catalogues. The basis of computation should be the grate surface and the rate of combustion. In house heating boilers of medium size not more than

5 pounds of coal should be burned per square foot of grate surface per hour. As to a 5-pound rate being a fair maximum to assume, it may be compared with horizontal tubular boiler practice in which, with easy firing, a 10 to 12 pound rate is common. Such boilers have 33 to 40 square feet of heating surface per square foot of grate, whereas common sizes of house heating boilers have, roughly speaking, 16 to 20. Hence, with half the heating surface the rate of combustion should be proportionally lower in order that the heat may be as well absorbed. This would give a 5 or 6 pound rate for house heaters.

HOW COMPUTE SIZE OF BOILER.

To ascertain the size of boiler necessary to supply a given amount of direct radiation, say, 1,500 square feet, for example, including the surface in mains, first multiply the total surface by the heat given off per square foot per hour. With hot water, in the case taken for illustration, this would be $1{,}500 \times 150 = 225{,}000$ heat units. Assuming 8,000 heat units to be utilized per pound of coal burned, each square foot of grate, with a 5 pound rate of combustion, will give to the water in the boiler 40,000 heat units per hour. Therefore the grate surface required will be $225{,}000 \div 40{,}000 = 5.62$ square feet.

RATE OF COMBUSTION.

The rate of combustion should not exceed 5 pounds for boilers having, say, not over 6 or 8 square feet of grate surface.

Boilers with two or three times as large a grate are generally cared for by a paid attendant, in which case there is no objection to burning coal at a faster rate. Such boilers generally have more heating surface in proportion to the grate than the smaller ones, hence the increased output of heat will be readily absorbed and the boiler will be just as economical as a smaller one burning coal more slowly.

Small boilers with 10 to 15 square feet of heating surface per square foot of grate should be rated to do their work on a 3 to 4 pound rate of combustion, corresponding to about 160 to 210 square feet of hot water radiating surface per square foot of grate.

Medium size boilers, with 16 to 20 square feet of heating sur-

face to 1 of grate, should be based on burning 4 to 5 pounds of coal on each square foot of grate per hour, corresponding, in round numbers, to 210 to 260 square feet of hot water radiation per square foot of grate.

Large size boilers with 21 to 25 or more square feet of heating surface per square foot of grate may be rated on a coal consumption of 6 to 7 pounds per square foot per hour, or even a trifle higher rate, where the heating surface is ample, corresponding approximately to 320 to 370 square feet of hot water radiation per square foot of grate. With steam radiation giving off, say, 250 heat units per square foot per hour, the same grate would carry only 150-250 = 3-5 as much surface as with hot water radiation.

The maximum night rate, when a boiler is expected to run at least eight hours without attention, should not exceed 4 pounds, equal to 32 pounds of coal burned on each square foot of grate in that length of time. With the 4-pound rate of combustion assumed, a fire one foot thick would burn about half through during the night, leaving an ample quantity of unconsumed fuel on the grate to readily ignite the fresh fuel added in the morning. With a higher rate of combustion a thicker fire would be necessary. Too great a depth, however, would interfere with the draft.

One of the essentials in a house heating boiler is a fire box of sufficient depth to permit carrying a good deep fire. Thin fires require too frequent attention. Avoid boilers with grates of excessive length, owing to the difficulty of properly handling the fire.

AMOUNT OF FUEL FOR A SEASON.

To compute the season's coal consumption in a house is, as heating men know, a very uncertain problem. The radiating surface or the grate area may be taken as a basis. If the boiler is properly proportioned for its work, so that the maximum rate of combustion need not exceed that stated above, the amount of coal required may be computed most readily by basing it directly on the grate surface. With a climate like that in many sections of the northeastern part of this country, where the heating season is

of about seven months duration and the average outside temperature during that time is not far from 40 to 45 degrees, the average rate of combustion will be, roughly, from 1¼ to 1¾ pounds per square foot per hour.

Take, for example, a boiler of medium size, in which the coal is to be burned no faster than a 5 pound rate in zero weather. Assume the heating season to last 200 days, or 4,800 hours. With an average outside temperature of, say, 45 degrees, the average rate of combustion, based on the difference between the indoor and outdoor temperatures, will be only $\frac{25}{70} \times 5 = 1.79$ pounds. Making allowance for the lower temperature maintained at night brings the average rate down to about 1.68 pounds. This, with a boiler having 4 square feet of grate surface, gives $4 \times 1.68 \times 4{,}800 = 32{,}256$ pounds, or about 16 tons for the season.

If the estimate be based on the radiating surface instead of on the grate area, we may assume, for example, a house heated by 1,000 square feet of direct radiation, including mains as a part of the surface.

Using the figures previously stated—viz., 150 heat units per square foot of direct hot water radiating surface and 8,000 heat units utilized per pound of coal, we have $1{,}000 \times 150 \div 8{,}000 = 18.8$ pounds per hour in coldest weather. The average hourly consumption, with an outside temperature of 45 degrees, would be $\frac{25}{70} \times 18.8$, and the total for the season of 4,800 hours $\frac{25}{70} \times 18.8 \times 4{,}800 =$ approximately 32,200 pounds. This would be reduced, owing to the lower temperature kept up at night to, say, 15 tons.

With indirect radiation, reduce to approximate equivalent direct radiation by multiplying by not less than 1.6. Some boiler manufacturers recommend multiplying by 1.75. Expressed in another way the computation just made, based on hot water radiation, gives about 40 pounds of coal per season per square foot of surface in radiators, allowing 25 per cent. for mains. With steam radiation the coal required would be $\frac{250}{150} \times 40 =$ about 70 pounds.

It may be well to repeat that the above computations apply only

to properly proportioned systems. If a boiler is known to be small for its work a higher average rate of combustion must be assumed and vice versa.

There is no economy in having a boiler so large that the fire must be checked by opening the feed door or running with a very low rate of combustion.

With a pair of boilers it is better to run one at its maximum rate until the second one is needed rather than run both with drafts checked nearly to the limit.

CHAPTER III.

HEAT GIVEN OFF BY RADIATORS AND COILS.

Repeated tests have shown the amount of heat given off by ordinary cast iron radiators per square foot of heating surface per hour per degree difference in temperature between the steam or water in the radiator and the air surrounding same to be about

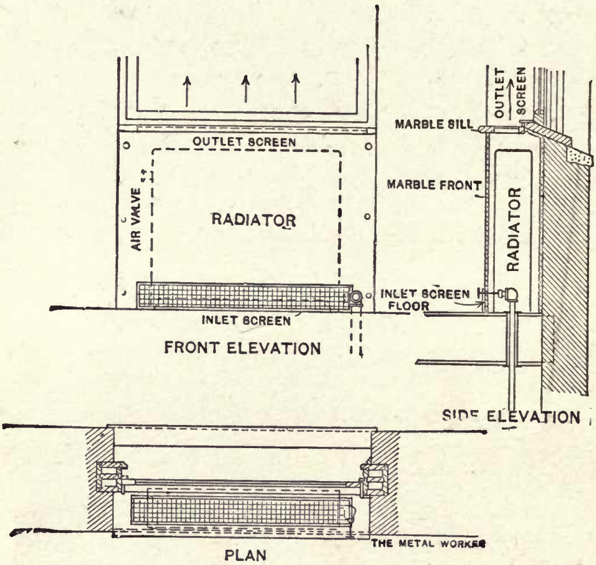

Fig. 16.—Plan and Front and Side Elevations, Showing Method of Concealing Radiator with Marble Wainscoting.

1.6 heat units. With this as a basis a steam radiator under 5 pounds pressure, corresponding to 228 degrees, surrounded by air at 70 degrees (neglecting the difference in temperature between the air near the top and the bottom of the radiator), will give off (228 — 70 degrees) $\times$ 1.6 heat units per square foot per hour = 253, commonly taken as 250. With hot water at an average temperature of 160 the heat given off is (160 — 70) $\times$ 1.6 = 144, commonly taken at 150.

These are good average figures to use. If we go into the subject closely we note that low radiators are more effective than high ones and those of single column pattern are more effective than deeper radiators, since they radiate their heat more freely and air will circulate around them to better advantage.

Wall radiators and coils of pipe are still more effective, overhead coils, with pipes side by side, giving off more heat per square foot than those on walls with the pipes one over the other. The advantage in the location of the latter, however, more than offsets the greater efficiency of those placed overhead, as is common in mill heating. Coils may be based on 300 to 350 heat units per square foot per hour with low pressure steam, and wall radiators on about the same amount. Concealed radiators, like the one illustrated in Fig. 16, give off practically no heat by radiation, but heat the room by heating the air passing over them—that is, by convection. Such radiators should therefore be rated to give off not more than 200 heat units per square foot per hour, depending on the hight and arrangement.

NOTES ON HEAT EMITTED BY DIRECT RADIATORS.

Professor Carpenter, in Vol. I, Transactions A. S. H. & V. E., states: "The capacity for heat transmission increases at a much higher rate than the difference of temperature. The efficiency of the radiator will be greatly increased by increasing the steam pressure or by forcibly bringing the air in contact with it. The heat emitted per hour under different conditions by the same radiator was found by tests to vary about 15 per cent., this variation being largely due to a difference in temperature and also to changes in velocity of air passing over the radiator.

"With radiators of the same form, but of different heights, the lower the radiator the more efficient. In the case of a Royal Union radiator 17 inches high, with practically the same amount of heating surface as another 37 inches high, 50 per cent. more heat was emitted by the low radiator. The radiator coefficient for a difference of temperature of 150 degrees is usually about 1.6 heat units; that for a 2-inch horizontal pipe 3.8 heat units; 1-inch pipe, 5.7 heat units.

"Radiators with one row of tubes are much superior to those of the same kind with two or more rows of tubes. The fact that

low radiators are more efficient than high ones would indicate that the tubes in the high radiators are too closely placed; that the air in its passage upward reaches nearly its maximum temperature in a short distance and from that point upward absorbs but little heat."

The average of many tests on ordinary cast iron radiators appears to confirm the figure of 1.6 heat units per square foot an hour per degree difference in temperature as a fair one to use.

Tests made by the City Engineer of Richmond, Va., on several types of radiators commonly used, gave results ranging from 1.43 heat units to 1.81 heat units per square foot an hour per degree difference in temperature. The average of the tests on five different makes of radiators was 1.68.

Monroe, in his book on "Steam Heating and Ventilation" states: "The writer found that under the conditions in his testing plant the 38-inch, 2-column cast iron radiator gave out 1.6 heat units per square foot per hour per degree difference of temperature, with an average difference of 147.5 degrees."

He states that "within the limits of ordinary radiator practice with steam temperatures from 212-230 degrees, and mean air temperatures from 40-70 degree, the coefficient of 1.6 will not vary more than 9 per cent. due to the difference in temperature between the steam and air. The radiator which has the most open space around its surface and the largest uninterrupted exposure to the surrounding air will give out the most heat per square foot under the same conditions. In compliance with this rule, other things being equal, narrow radiators are more effective than wide ones and low ones than high ones. Professor Cooley found that a single coil of horizontal pipes set side by side gives out 40 per cent. more heat per square foot than a two column cast iron radiator under the same conditions."

In the "Plumbers' and Fitters' Pocket Book," published by the International Correspondence Schools, 1905, a statement is made that the heat units emitted per hour per square foot of surface per degree difference in temperature amounts with 90 degrees difference (which would correspond approximately with hot water heating conditions), to 1.41 for radiators 40 inches high, 1.7 for radiators 24 inches high, 1.62 for single column radiators

40 inches high, and 2.22 for those 24 inches high. The figures taken in the same order for over 160 degrees difference in temperature corresponding practically to steam heat conditions would be 1.66, 1.98, 1.88 and 2.59.

The Fowler & Wolfe Mfg. Co.'s catalogue gives a summary of tests as follows:

TABLE II.

Summary of Tests of various steam radiators made at Sibley College, Cornell University, by Messrs. Camp, Woodward and Sickles, mechanical engineers, under the direction of R. C. Carpenter, M.S.C.E., M.M.E. (This summary is the average of several consecutive tests made on these several radiators.)

These tests were all made in the same closed room under even temperatures and under same conditions.	F. & W. wall radiator. Standard 7-foot section tested.	Standard hight 3-column cast iron radiator.	A standard hight cast iron radiator with loops attached to base.	A standard hight radiator made of 1-inch wrought iron pipe attached to cast iron base 3 rows wide.	A standard hight cast iron 2-column radiator.
*B. T. U. heat radiated per hour per square foot of actual surface. Per degree difference in temperature......	2.325	1.732	1.705	1.643	1.319
B. T. U. heat radiated per hour per rated square foot of surface. Per degree difference in temperature	2.400	1.712	1.594	1.266	1.266
Steam condensed per hour per actual square foot of heating surface. Pounds.	0.351	0.236	0.239	0.182	0.182

* B.T.U. = British thermal units, or heat units.

Reference is made in the Heine Safety Boiler Co.'s catalogue to the average of four experiments on the condensation in uncovered pipes which showed with an average steam pressure of 5 pounds gauge, 2.236 heat units per square foot per hour per 1 degree F. Other tests showed a loss of 2.812 for bare pipe.

Mr. A. R. Wolff gives 250 heat units per square foot per hour for ordinary cast iron radiators with steam from 3 to 5 pounds per square inch, and recommends about 60% of this amount for hot water heating.

The results of a number of radiator tests are given in Mills's book on "Heating & Ventilation," Vol. II., page 335. The heat

emitted from cast iron radiators, according to these tests, ranges from 1.4 for certain types of cast iron radiator, to 2.38 for single column wrought iron tube radiator. Heat given off by horizontal pipes is as follows: 1-inch pipe 2.73; 2-inch pipe 2.3; 3-inch pipe 2.33.

The following figures are taken from "Steam in Covered and Bare Pipes," by Paulding:

TABLE III.

LOSS OF HEAT FROM PIPES.

Name of experimenter.	Size of pipe. Inches.	Temperature of steam. Deg. F.	Temperature of air. Deg. F.	B. T. U. per square foot per hour per 1 deg.
Barrus	2	325.2	56.6	3.01
Barrus	2	365.4	63.2	3.25
Barrus	10	365.3	73.6	3.18
Hudson-Beare	3.53*	358.0	67.0	3.10
130 pounds	2	354.7	80.1	3.13
Jacobus	2	300.6	71.2	2.78
Brill	8	344.5	75.5	2.71

* Actual outside diameter.

Since the heat given off is roughly proportional to the difference in temperature between the steam and the air in the room, radiators placed in rooms to be heated to a temperature lower than 70 degrees, say 50 degrees, will give off with radiators at 228 degrees $\frac{(228-50)}{(228-70)} \times 250$ heat units = about 280 heat units. In this connection it may be well to remark that in computing boiler capacity one must remember that catalogue ratings are based on the radiators being placed in rooms at 70 degrees. The radiation must be reduced to equivalent surface when surrounded by air at 70 degrees temperature.

It has just been shown that in rooms at 50 degrees the radiators give off 280 heat units, against 250 heat units in 70-degree rooms; hence, a boiler rated for, say 2500 square feet will carry only $\frac{250}{280} \times 2500 = 2230$ square feet if the rooms are to be heated to only 50 degrees.

HEAT GIVEN OFF BY INDIRECT RADIATORS.

Indirect radiators of the pin or similar type, with extended surface, arranged somewhat as shown in Fig. 17, give off heat not

only in proportion to the difference in temperature between the steam and the surrounding air, but in proportion (though not directly) with the volume of air coming in contact with them.

The tests made some years ago by John H. Mills have been frequently quoted by writers on heating and ventilation. The writer has reduced these tests to a zero basis for the entering air, the data being given in the following table.

TABLE IV.

THE HEAT UNITS GIVEN OFF PER SQUARE FOOT PER HOUR FROM INDIRECT PIN RADIATORS HAVING 40 PER CENT. PRIME SURFACE.—STEAM, 5 POUNDS PRESSURE; ENTERING AIR, 0 DEGREE F.

Cubic feet of air per hour passing over each square foot of heating surface.	Heat units given off per hour per square foot of extended surface.	Velocity in feet per minute between 10 square foot sections, having 1/3 square foot air space between each two sections.
100	370	50
200	540	100
300	700	150
400	850	200
500	1,015	250
600	1,175	300
700	1,330	350
800	1,500	400

It is common to assume about 400 heat units to be given off per square foot an hour from ordinary indirect pin radiators with low pressure steam. Short vertical flues mean low velocities; higher ones give an increased air flow.

The table shows that where a good velocity between the sections may be secured their effectiveness is increased and less surface is required.

COMPUTING INDIRECT RADIATING SURFACE.

To illustrate the use of the table, suppose we have a room 20 × 30 × 12 which it is desired to heat by indirect radiation and change the air every 12 minutes—contents equals 7200 cubic feet. With 5 changes per hour 36,000 cubic feet must be supplied. The heat loss by transmission, with two sides exposed, would be about 24,000 heat units per hour. The loss by ventilation would be 36,000 × 1¼ (1¼ representing the heat units carried away by each cubic foot of air escaping from a 70-degree room, with outside air at 0 degree) = 45,000. Adding these, the total heat loss is 81,000 heat units per hour. Assuming 400 heat units per

square foot of radiation per hour gives a trifle over 200 square feet of surface, or a ratio of 1 to 36 cubic feet. With 36,000 cubic feet per hour supplied the air admitted to the indirect radiators would be 36,000 ÷ 200 = 180 cubic feet per square foot (neglecting the difference in volume between air at 70 degrees and at 0 degree). The table shows that with 200 cubic feet per square foot per hour 540 heat units are given off; hence we should expect

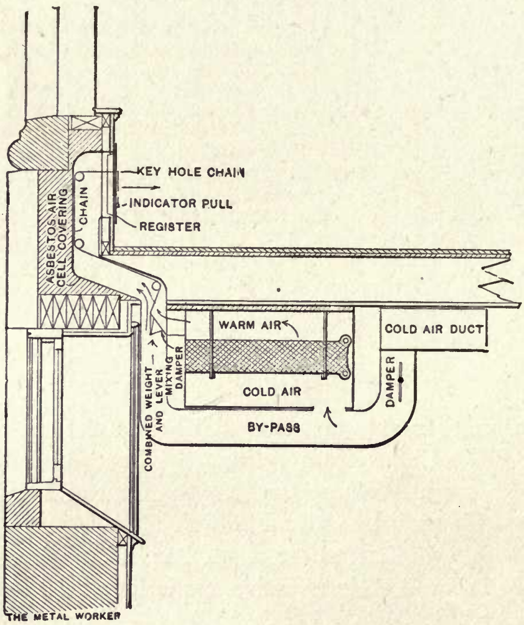

Fig. 17.—Indirect Radiator Connections.

that with 180 cubic feet about 500 heat units in round numbers would be given off, in which case only 81,000 ÷ 500 = 162 square feet would be necessary. One must always be certain that the air space through the groups of radiators is considerably in excess of the area of flues connected therewith. The rule to allow 2 square inches of flue area to the first floor, 1½ to the second floor and 1¼ to the third and fourth floors is simple, and gives good results in dwelling house work when the radiation is properly proportioned. That is just the difficulty, however, for in case of a mistake in the radiation a second mistake follows in the flues.

Taking $1\frac{1}{2}$ square inches of flue area per square foot of indirect radiating surface as a fair average for a house, a bench or stack of 100 square feet would have flues aggregating 150 square inches. The flue area between the sections would be about 480 square inches, or over three times the flue area; thus, common practice dictates that the velocity between the sections of pin radiators shall be only about one-third that in the flues. The rule to make the indirect surface 50 per cent. more than the direct radiation that would be required may be shown on a heat unit basis to be very nearly true under certain conditions. For example,

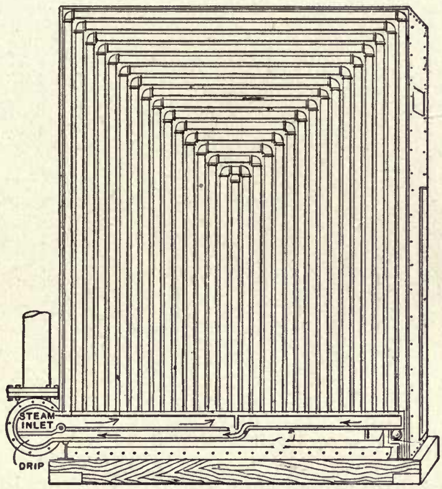

Fig. 18.—Blower System Heater.

take a corner room $16 \times 20 \times 10 = 3200$ cubic feet, the heat loss from which is 14,400 heat units per hour. With direct steam radiation rated at 250 heat units $14{,}400 \div 250 = 58$ square feet would be required. Now, to heat the same room by indirect radiation at 400 heat units per square foot, the air to enter the room at 120 degrees, with 0 degree outside, about 86 square feet would be required, computed as follows:

One cubic foot of air at 120 degrees weighs 0.068 pound. Its specific heat is 0.238, therefore the heat units brought in by a cubic foot of air at 120 degrees is $0.068 \times 120° \times 0.238 = 1.94$.

Of this only $\frac{50}{120}$ is available to offset the loss of heat by transmission, the other $\frac{70}{120}$ escaping with the air leaking out at 70 degrees temperature. $\frac{50}{120} \times 1.94 = 0.810$ heat unit. To make good the loss of 14,400 heat units per hour by transmission $14,400 \div 0.810 = 17,800$ cubic feet of air per hour at 120 degrees must be supplied. Each cubic foot brings in 1.94 heat units; total equals $17,800 \times 1.94 = 34,500$ heat units, which divided by 400

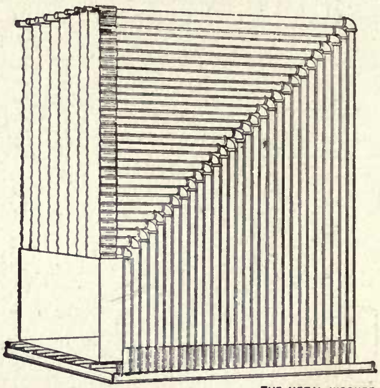

Fig. 19.—A Heater for Blower Use.

gives 86, an amount almost exactly 50 per cent. in excess of the direct radiation required.

HEAT GIVEN OFF BY HEATERS COMBINED WITH FANS.

It is not uncommon to secure an emission of 1500 to 2000 heat units or more per square foot of pipe coils when zero air is entering the heater at a velocity of 1000 to 1200 feet per minute, measured between the pipes and the steam is 2 to 5 pounds gauge pressure. See Figs. 18 and 19.

The heat given off per square foot by supplementary heaters or reheaters, as shown in Fig. 20, with which air at, say, 50 to 70 degrees from the main tempering coils comes in contact would be not far from 1000 to 1200 heat units in the case of low pressure steam. The velocity of the air and the depth of heaters—

that is, the number of coils of pipe they contain—have much to do with their efficiency, which depends chiefly on the steam pressure. Assuming a main tempering coil arranged to have the air blown through it by a fan or blower, as in Fig. 21, or to have the air drawn through, as shown in Fig. 22, to give off 2000 heat units per square foot per hour, what amount of surface would be necessary to raise the temperature of 30,000 cubic feet per minute 70 degrees from zero?

Since one heat unit will raise the temperature of approximately 50 cubic feet of air from 0 degree through 1 degree, to

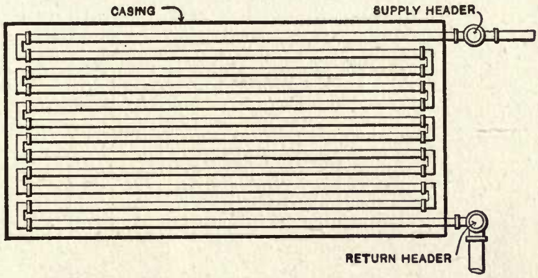

Fig. 20.—Supplementary Heater or Reheater.

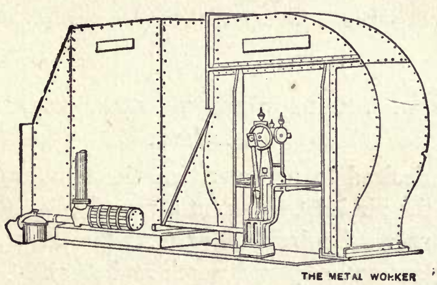

Fig. 21.—A Fan Blowing Air through Heater.

raise $30{,}000 \times 60 = 1{,}800{,}000$ cubic feet per hour 70 degrees, would require $\frac{1{,}800{,}000 \times 70}{50} = 2{,}520{,}000$ heat units, which could be obtained by using a heater of $\frac{2{,}520{,}000}{2000} = 1260$ square feet, or about 3600 lineal feet of 1-inch pipe.

CAST-IRON RADIATION FOR USE WITH FANS.

Cast-iron radiation made up in large sections and of a design to give prime surface in place of the extended surface so long in vogue with indirect radiation is now largely used in connection with fans.

These sections are more easily handled than large coils and may be arranged in a variety of ways to suit the space at one's disposal. Manufacturers' catalogs relating to this product give an unusual amount of valuable engineering data.

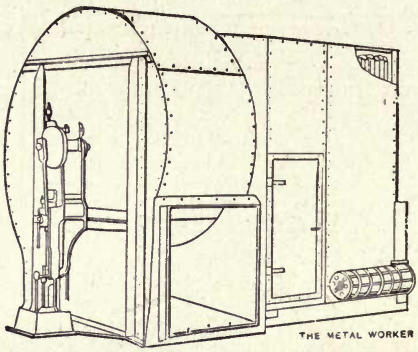

Fig. 22.—Fan or Blower Drawing Air through Heater.

TEMPERATURE OF AIR REQUIRED TO HEAT ROOMS BY INDIRECT RADIATION.

It may be desired to predetermine the temperature that must be secured at the air inlet to warm a room.

Take a corner schoolroom, for example, 28 x 32 x 12, with 30 per cent. glass and exposed north and west.

The equivalent glass surface, rating the wall as equivalent to one-quarter as much actual glass surface, will be 342 square feet.

The heat lost through same per hour will be $342 \times 85 \times 1.25 = 36{,}340$ heat units. (1.25 being the factor for N. or W. exposure.)

With the standard air supply to a 50-pupil room of 1500 cubic

feet per minute the loss of heat by leakage—that is, by the removal of air through the ventilating flues—will be $60 \times 1500 \times 1\frac{1}{4}$ (since $1\frac{1}{4}$ heat units are removed by each cubic foot of air escaping from a room at 70 degrees when the outside temperature is at 0 degree) $= 112{,}500$ heat units per hour. Total heat loss equals 148,840. To make good the loss of heat through walls and glass the 90,000 cubic feet of air per hour supplied to the room (the volume being based on 70-degree temperature) must be superheated above the room temperature an amount equivalent to the 36,340 heat units transmitted through walls and glass.

The weight of 90,000 cubic feet of air at 70 degrees is about $90{,}000 \times 0.075 = 6750$ pounds. The specific heat of air is 0.238—that is, one heat unit will raise the temperature of about 4 pounds of air 1 degree.

Therefore, 36,340 heat units would raise the temperature of $36{,}340 \times 4 = 145{,}360$ pounds of air 1 degree, or would raise the temperature of 6750 pounds of air; $145{,}360 \div 6750 =$ about 22 degrees.

That is, the air would have to be superheated at least 22 degrees above the room temperature of 70 degrees to maintain the room at that temperature under the conditions stated—viz., with a change of air about every eight minutes. As a matter of fact, with the indirect system there is a considerable difference between floor and ceiling temperatures in high studded rooms, which means that if 70 degrees is to be maintained near the floor a considerably higher temperature must be maintained above, with a consequent increase in the loss of heat by transmission; therefore, instead of 92 degrees, as above computed, based on an average temperature at walls of 70 degrees, the inlet temperature would probably have to be kept at not less than 100 degrees in zero weather, especially if the windows were not tightly fitted.

SIZE OF ASPIRATING HEATERS OR COILS.

To compute the size of heaters or coils to be placed in ventilating flues, as shown in section and elevation in Figs. 23 and 24, to produce an aspirating effect in a system of ducts, as shown by plan and elevation in Figs. 25 and 26, we may proceed as follows:

Suppose it is desired to remove 3000 cubic feet of air per minute from a room. Knowing the size and hight of the flue, for example, 10 square feet area and 40 feet high above where the coil

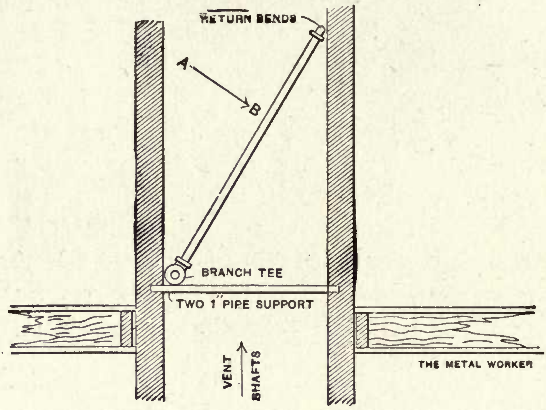

Fig. 23.—Section through Vent Flue, Showing Aspirating Coil.

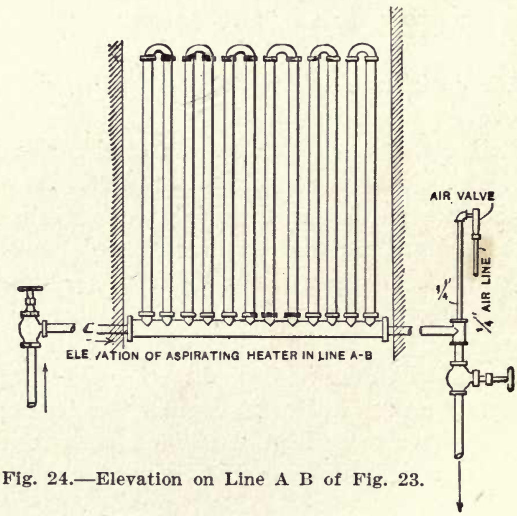

Fig. 24.—Elevation on Line A B of Fig. 23.

is to be placed, look up the flue velocities in Table XVII—the excess of temperature over that outdoors that must be maintained in the flue to produce the required velocity. In this case the velocity must be $3000 \div 10 = 300$ feet per minute, and the excess temperature required, taken from Table XVII, is 20 degrees.

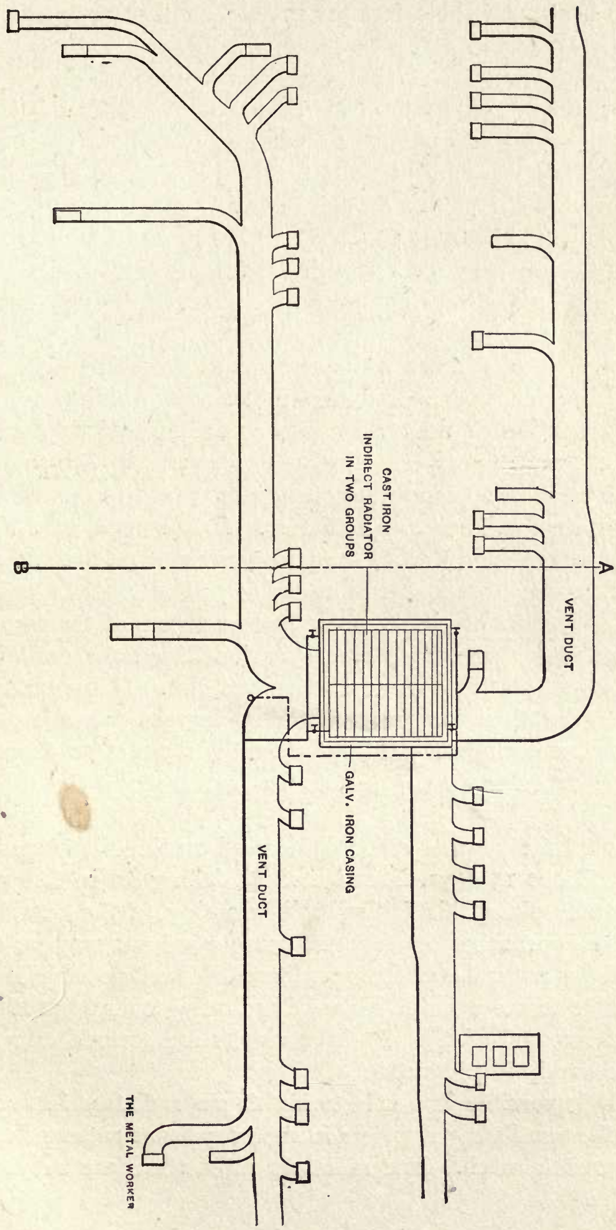

Fig. 25.—Plan of Ducts Connected with an Aspirating Heater.

To heat 3000 cubic feet per minute 20 degrees would require $\frac{3000 \times 60 \times 20}{55} = 65{,}454$ heat units per hour (55 representing the number of cubic feet of air heated 1 degree by 1 heat unit).

With an aspirating heater made up of ordinary pin radiators, giving off, say, 400 heat units per square foot of extended surface per hour, and this would be a fair allowance, the surface required would be $65{,}454 \div 400 = 163$ square feet. The sections should be coupled together with extra long nipples.

One should always compute the air space through heaters to make sure it is ample.

The free area between the sections of the heater should be at least 20 per cent. greater than the flue area, to allow for the increased friction of the air in passing over the pins or extended surface. A temperature rise of 20 degrees in the ventilating flues to produce an aspirating effect would require the use of very large heaters and coils, or radiators. It is therefore seldom that a temperature rise of more than 10 degrees is provided for.

This means that a 40-foot vent flue proportionel to handle the required volume of air with a 20-degree excess of temperature in the flue over that outdoors will work without the assistance of a coil up to 50 degrees outside temperature. If the outdoor air is 60 degrees then 10 degrees of the 20 degrees excess is provided by the air entering the vent flue from the room at 70 degrees, the balance, or other 10 degrees, to be furnished by the aspirating coil.

Should the weather reach 65 degrees outside the excess in the flue would be 15 degrees, and a slight falling off in flue velocity would take place, this falling off increasing as the outside temperature approaches 70 degrees, when windows may be opened and ample natural ventilation secured. When possible it is far preferable to use a fan in place of aspirating coils to produce the desired removal of air. Positive results are secured and the air may be handled at less cost.

It is impossible as a rule to install pin radiators in flues just above the ventilating registers in rooms without cutting down the flue area too much. The radiators must therefore be placed in the attic.

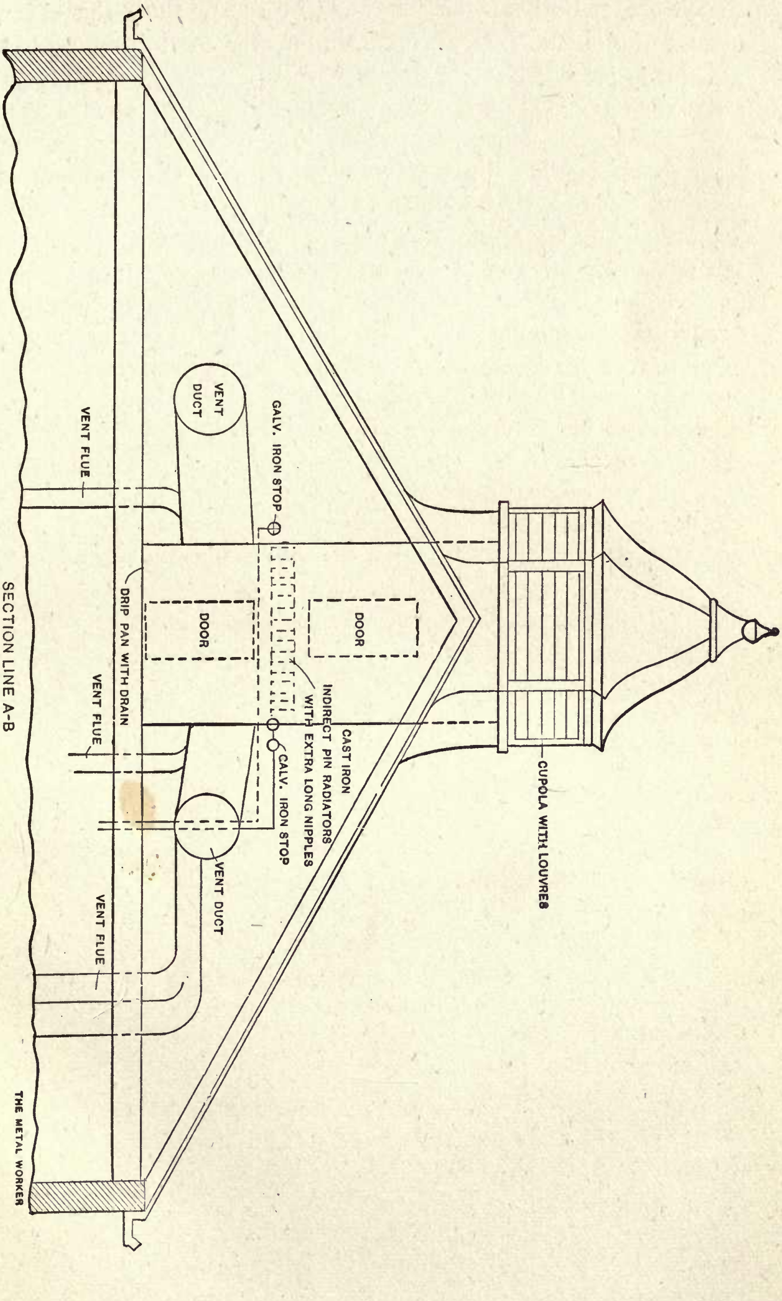

Fig. 26.—Elevation of System on Line A B of Fig. 25.

More radiation must be used, however, since the chimney effect of the flue is decreased the nearer the top the aspirating heater is placed.

TABLE V.

A table giving the approximate velocity of air in flues of various heights will be found on page 71.

Where small volumes of air are to be removed coils or lines of pipes may be used to advantage in place of cast iron radiators, care being taken not to block off too much of the flue area. Such coils may be computed on a basis of 600 heat units or more per square foot per hour, depending on the flue velocity.

CHAPTER IV.

THE LOSS OF HEAT BY TRANSMISSION, COMPUTING RADIATION, HORSE POWER REQUIRED FOR HEATING.

The following tables have been computed from data presented in a series of articles on "German Formulas and Tables for Heating and Ventilating Work," by Prof. J. H. Kinealy, published in bound form, pocket size, $1.00. The values given include those for a greater variety of building materials than the writer has seen published elsewhere. The values for glass and brick work agree pretty closely with those commonly used in this country.

TABLE VI.

LOSS OF HEAT THROUGH BRICK WALLS OF APPROXIMATELY THE THICKNESS STATED.

70 degrees inside, 0 degree outside.

Thickness of wall, inches	8	12	16	20	24	30	36
Heat units per square foot per hour	24	21	18	16	14	12	10

Tables showing the relative transmitting power of solid brick walls and those with air spaces about 2.4 inches wide show that those with the air space transmit about 15 to 20 per cent. less heat than the solid walls. This applies only to walls, say, 8 to 16 inches thick. With thicker walls the saving due to an air space is much less.

TABLE VII.

LOSS OF HEAT THROUGH STONE WALLS, RUBBLE OR BLOCK MASONRY.

70 degrees inside, 0 degree outside.

Thickness of wall, inches	12	16	20	24	28	36	44
Heat units per square foot per hour	31	27	25	21	19	17	14

The values given are for sandstone; about 10 per cent. should be added for limestone.

TABLE VIII.

LOSS OF HEAT THROUGH PINE PLANKS.

70 degrees inside, 0 degree outside.

Thickness of planking, inches	1½	2	2½	3
Heat units per square foot per hour	21	18	16	14

* A. R. Wolff's values for brick walls closely approximate those given except for 8-inch wall, which he states is 29, and for 4-inch wall is 46 B. T. U. per sq.ft. per hour 70 degrees difference.

TABLE IX.

LOSS OF HEAT THROUGH WINDOWS AND SKYLIGHTS AND THROUGH OUTSIDE WALLS OF FRAME CONSTRUCTION, EXPRESSED IN HEAT UNITS PER SQUARE FOOT OF EXPOSED WALL PER HOUR.

70 degrees inside, 0 degree outside.

	Heat units per square foot per hour.
Single window	77
Single window, double glass	43
Double window	32
Single skylight	81
¾-inch sheathing and clapboards	20
¾-inch sheathing, paper and clapboards	16

Professor Kinealy states: "These can hardly be considered much more than rough approximations on account of the uncertainty due to leakage."

TABLE X.

LOSS OF HEAT, EXPRESSED IN HEAT UNITS PER SQUARE FOOT OF SURFACE PER HOUR, THROUGH PARTITIONS, FLOORS AND CEILINGS SEPARATING WARM ROOMS AT 70 DEGREES FROM COLD ROOMS AT 40 DEGREES.

	Heat units.
Ordinary stud partition, lath and plaster one side only	18
Ordinary stud partition, lath and plaster both sides	10
Ordinary lath and plaster ceiling separating unheated space from heated rooms	18
Floor, single, thickness ¾ inch, warm air above and cold space below:	
(*a*) No plaster beneath joists	13
(*b*) Lath and plaster beneath joists	8
Floor, double, thickness 1½ inches, warm room above and cold space below:	
(*a*) No plaster beneath joists	9
(*b*) Lath and plaster beneath joists	5

The heat losses stated in the tables are to be increased as follows, based on the practice of different German engineers:

TABLE XI.

	Per cent.
For northeastern, northwestern, western or northern exposure	20 to 30
For rooms 13 to 14½ feet high	6½
For rooms 14½ to 18 feet high	10
When the heating is continued during the day only	10
When the building is allowed to become thoroughly chilled at night	30
When the building remains for long periods without heat	50

COMPUTATION OF HEAT LOSSES AND RADIATION.

To illustrate the use of the German values given, suppose it is desired to compute the amount of steam radiation required to

heat a corner room 14 x 16 x 10 feet, exposed to the north and west, located below a heated room and over an unheated room; floor to be double, with under side of floor joists lathed and plastered; outside walls 12 inches, brick; glass 20 per cent. of the exposure, equal to 60 square feet, net wall equaling 240 square feet:

Heat losses:	
Wall, 240 × 21 heat units	5,040
Glass, 60 × 77 heat units	4,620
Total	9,660
Heat loss × exposure factor = 9,660 × 1.25	12,075
Heat loss through floor, 224 × 5	1,120
Total heat loss	13,190

Direct radiating surface is equal to the total heat loss divided by heat given off per square foot of radiating surface—viz.: 250 heat units, or 13,190 ÷ 250 = 53 square feet, giving a ratio of about 1 square foot to 43 cubic feet of space. It will be noted that no allowance has been made in the above example for air leakage. Professor Kinealy points out that the German engineers appear to make no allowance for this item, except as taken into account by the percentage addition for exposure. Some engineers in this country allow for the accidental leakage by assuming a certain rate of air change, say once an hour, for all rooms.

In large rooms having little exposure in proportion to the contents the loss of heat due to leakage, based on an hourly rate, is often as great as that through the walls, if not greater, which would call for more radiation than is found necessary in practice.

The question of leakage is an important one and requires good judgment for its proper determination. In preference to making a fixed allowance for leakage, based on the cubic contents, the writer has found it more satisfactory to consider the leakage to be sufficiently allowed for by the exposure factors of 1.25 for north or west and 1.15 for east, especially when using factor 77 or 85 for glass, and to make a separate allowance for the effect of the cubic contents on the heating of a room by adding to the loss of heat by transmission an amount of heat equal to the cubic contents in feet multiplied by 1/3 for room with two exposures, and the cubic contents multiplied by 2/3 for rooms with one exposure. This allowance will be found sufficient to provide for

reheating where the rooms are allowed to become somewhat chilled at night.

The reason for making a greater allowance for reheating in the case of rooms with one exposure than of those with two exposed walls is that the rate of transmission is somewhat greater per square foot through the single exposed wall having three partition walls radiating heat to it than through the same wall area of a corner room having only two interior walls radiating heat to the outer ones.

Furthermore the three inside walls on account of their greater surface, require more heat to warm them in a given time than do the two inside walls of a corner room of the same size, therefore, in order that corner rooms and single exposure rooms shall heat in approximately the same time, a greater allowance for reheating should be made for the latter.

COMPUTING DIRECT RADIATION ON THE HEAT UNIT BASIS.

Perhaps the most time consuming operation in connection with the work of the heating engineer or contractor is the computation of radiating surface. Innumerable rules have been devised, good, bad and indifferent, but the subject appears to have simmered down to the simple proposition that if the wall and glass surface and the required air change are known the heat losses due to transmission and leakage may be readily determined, and this total divided by the heat given off per hour per square foot of radiating surface gives the amount of radiation required.

The German values for the heat transmitting power of various substances of different thicknesses have been widely used since they were first introduced by A. R. Wolff. Tables or charts giving these values may be found in Kent's "Mechanical Engineers' Pocket Book" and in many trade catalogues. Values closely approximating these have been stated. The French values, based on the investigations of Péclet and introduced by Professor Carpenter, are in some cases considerably lower than those just mentioned, his value or coefficient for glass being 70, Wolff's being 85. Furthermore, Carpenter assumes a certain air change per hour by leakage in rooms heated by direct radiation, whereas Wolff provides for this loss by adding a certain percentage to

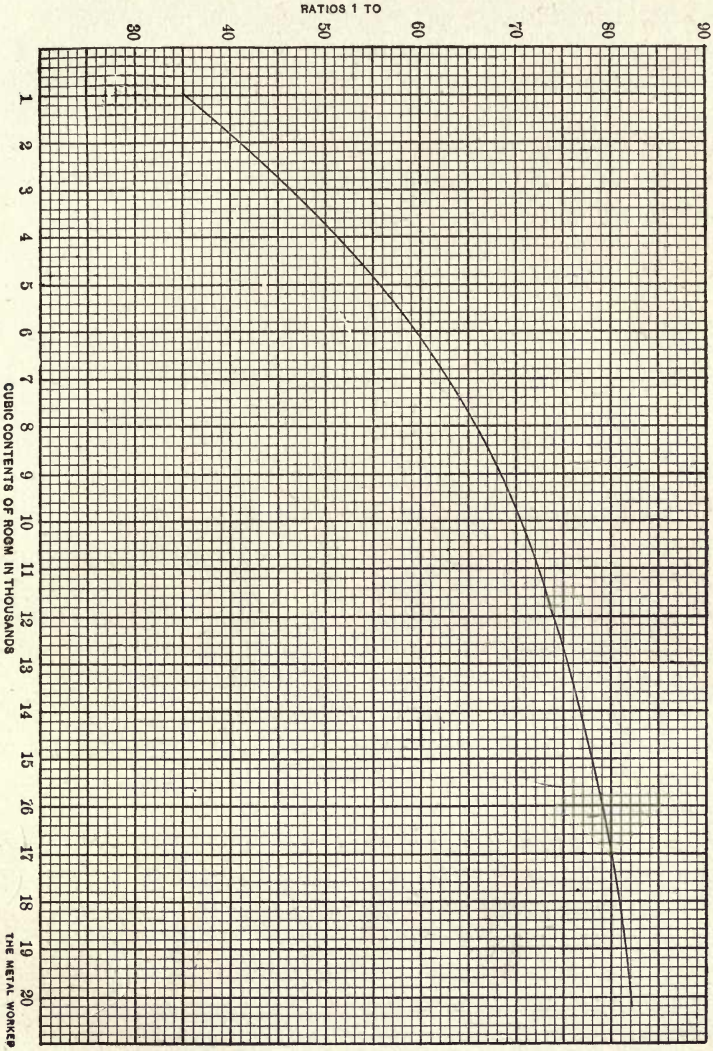

CHART I.

Ratios for DIRECT STEAM Radiating Surface in Rooms with TWO SIDES EXPOSED Toward the North and West, with Glass Surface Aggregating 20 Per Cent. of Total Exposure.

For northeast corner rooms use ratio 5 per cent. greater than given by chart.
For southwest corner rooms use ratio 10 per cent. greater than given by chart.
For southeast corner rooms use ratio 15 per cent. greater than given by chart.

the heat losses through walls and glass. Of course, where the leakage is great, as in rooms provided with ventilating flues, it is allowed for independently.

Admitting that the wall and glass surface affords the best basis on which to compute the radiating surface, it frequently happens in a contractor's office that insufficient time is given in which to lay out the work on this basis and prepare a bid. In house heating work especially some shorter method must often be used for the reason stated. In such cases an experienced man may be able to hit pretty close to the mark by "thumb rule," but, while quick, this method is a rather rough one.

Some simple method that will give reasonably accurate results that may be quickly arrived at is needed by many contractors. The author prepared, and has for several years used, the accompanying charts, Nos. 1 and 2, for computing direct steam and 3 and 4 for direct hot water radiation, the curves representing the mean or average of the German and French values with these modifications:

To the heat loss through walls and glass, based on German values, has been added a certain amount to allow for reheating the air in the rooms in case they should become chilled.

To the heat losses by transmission, computed on the French basis, has been added an amount representing the heat units escaping by a leakage of air equal to the contents of the room once each hour.

The tables are based on a glass surface equal to 20 per cent. of the total exposure. This the author has found to be a fair allowance; some rooms may have more than this amount, but an excessive glass surface is readily detected in inspecting plans and may be allowed for by adding to the radiating surface given by the chart an amount of radiation equal to about one-third of the excess of glass surface over the 20 per cent. on which the charts are based.

For example: If the total exposure is 400 square feet and the glass surface 120 square feet, or 40 square feet in excess of the glass surface based on 20 per cent. of the exposure, 13 square feet, being one-third of 40, should be added to the radiation computed by the chart.

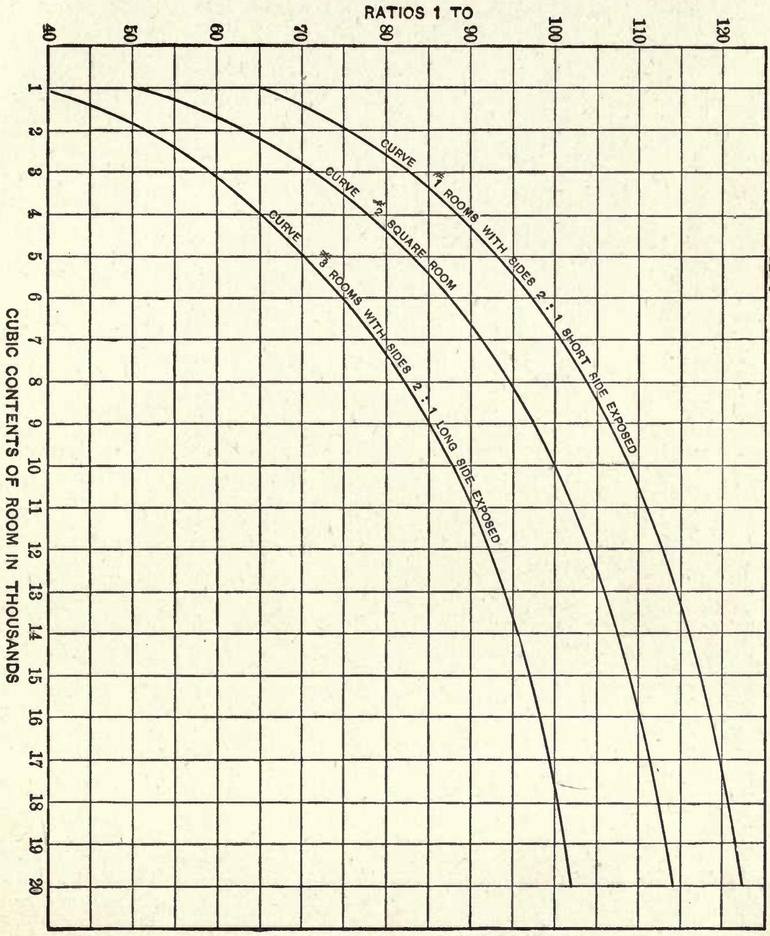

CHART 2.

Ratios for DIRECT STEAM Radiating Surface in Rooms Having Only ONE SIDE EXPOSED Toward North or West, with Glass Surface Aggregating 20 Per Cent. of Total Exposure.

Curve 1 is for rooms having length to width as 2 to 1, with short side exposed.

Curve 3 is for rooms having length to width as 2 to 1, with long side exposed.

Curve 2 is for square rooms with one side exposed.

For rooms with east, south or southeast exposure use ratio 10 per cent. greater than chart.

For rooms with southwest or northeast exposure use ratio 5 per cent. greater than chart.

Chart No. 1 shows a curve from which the proper ratios of steam heating surface to cubic contents may be determined for rooms with two exposed sides. The curve was computed for square rooms. Rectangular rooms of good proportions, however, have but little more exposed wall surface in proportion to their contents, and unless they are unusually long and narrow the ratio given by the chart may be safely used. The contents expressed in thousands of cubic feet is stated on the lower line and the ratio of radiating surface to contents is given in the vertical line at the left of the chart.

Example: What radiating surface should be used in a corner room 16 x 19 x 10 feet, having 3040 cubic feet? Just to the right of the 3000 line is a point representing the contents of 3040 cubic feet. Note where a line drawn vertically through this point would intersect the curve. In the left hand column this point of intersection is the ratio sought. The ratio in this case is about 1 to 47. The contents (3040) divided by this ratio gives 63 square feet of direct radiating surface.

Chart No. 2, for rooms with one exposure, contains three curves, one for rooms with sides in the proportion of 2 to 1 (24 x 12 feet, for example), having the long side exposed; one for similar rooms with the short side exposed and one for square rooms. Obviously it makes a great difference whether the long or the short side of a room is exposed.

For rooms having sides in the proportion of 1½ to 1 (15 x 10 feet, for example), with the long side exposed, compute the contents and proceed as explained in connection with Chart No. 1, selecting a point in Chart 2 midway between curve 2 and curve 3 on the vertical line corresponding with the contents. The proper ratio will be found in the left hand column opposite this midway point.

With rooms like the one described, but having the short side exposed, select a point midway between curve 1 and 2.

Example: What amount of steam radiating surface is required in a room 12 x 18 x 10 feet, having the 12-foot wall exposed? Contents 2160 cubic feet. On Chart 2 follow up the line representing the contents to a point midway between curves 1 and 2, then out horizontally to the left hand column. The ratio there found is

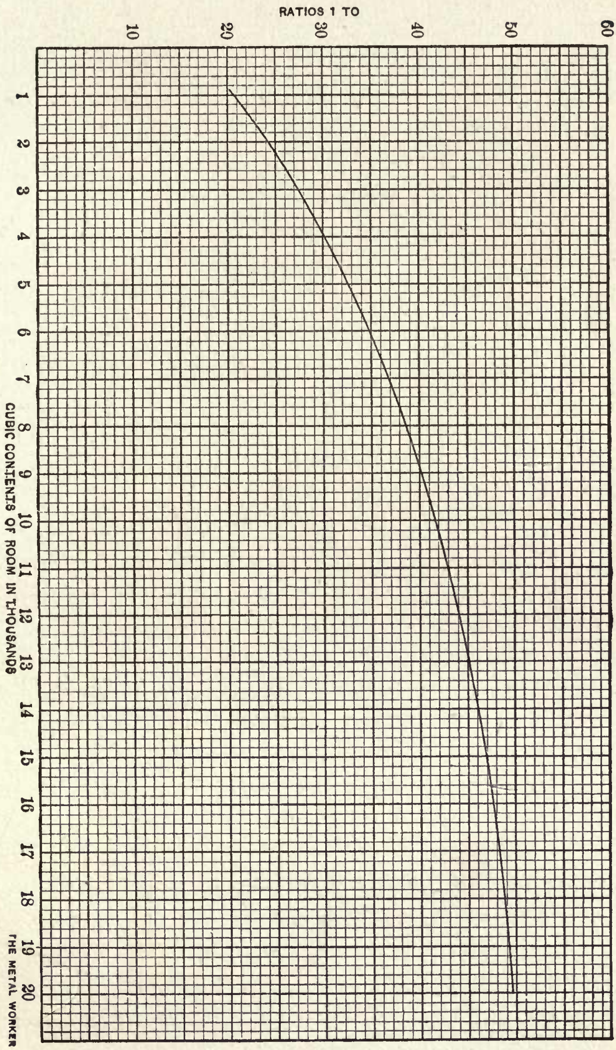

CHART 3.

Ratios for DIRECT HOT WATER Radiating Surface, Open Tank System, in Rooms with TWO SIDES EXPOSED Toward the North and West, with Glass Surface Aggregating 20 Per Cent. of Total Exposure.

For northeast corner rooms use ratio 5 per cent. greater than chart.
For southwest corner rooms use ratio 10 per cent. greater than chart.
For southeast corner rooms use ratio 15 per cent. greater than chart.

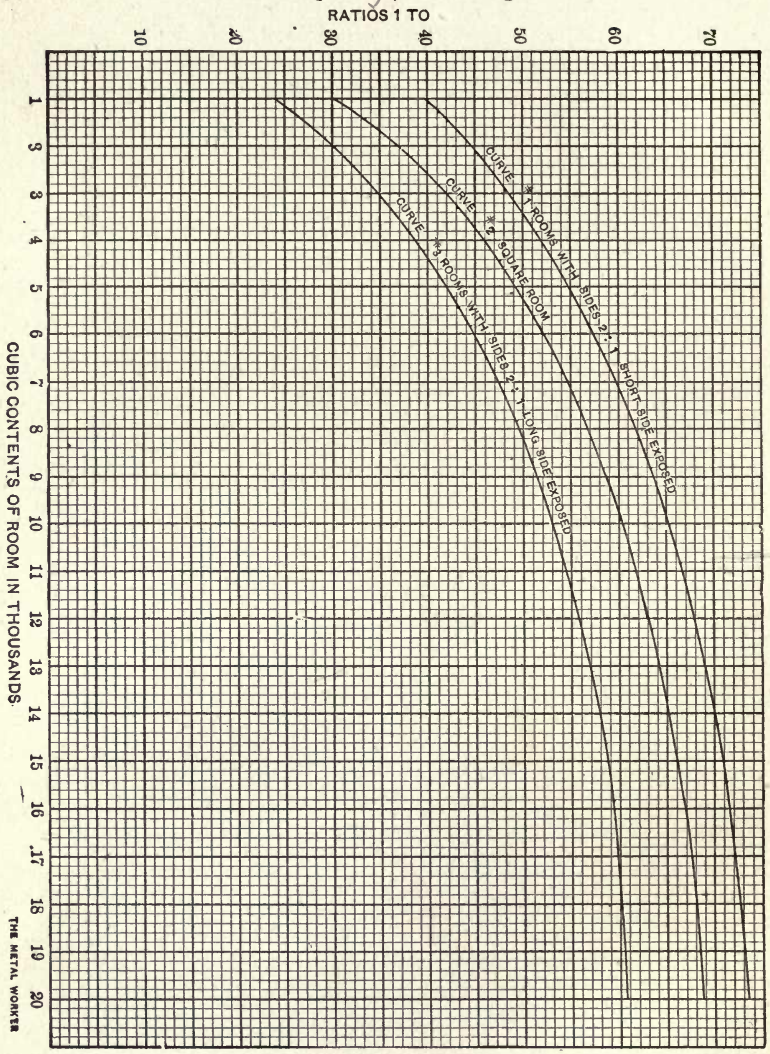

CHART 4.

Ratios for DIRECT HOT WATER Heating Surface, Open Tank System, in Rooms with Only ONE SIDE EXPOSED Toward the North or West, with Glass Surface Aggregating 20 Per Cent. of the Total Exposure.

Curve 1 is for rooms having length to width as 2 to 1, with short side exposed.

Curve 3 is for rooms having length to width as 2 to 1, with long side exposed.

Curve 2 is for square rooms, with one side exposed.

For rooms with east, south or southeast exposure use ratio 10 per cent. greater than chart.

For rooms with southwest or northeast exposure use ratio 5 per cent. greater than chart.

about 1 to 72, and the radiating surface $2160 \div 72 = 30$ square feet.

With this explanation of charts No. 1 and No. 2 for steam heating, the use of charts No. 3 and No. 4 for hot water heating will be readily understood without further examples.

THE BOILER HORSE-POWER AND RADIATING SURFACE REQUIRED TO HEAT ISOLATED BUILDINGS.

It is of interest to compute on a heat unit basis the boiler horse-power necessary to heat buildings under the conditions stated in connection with Chart 6.

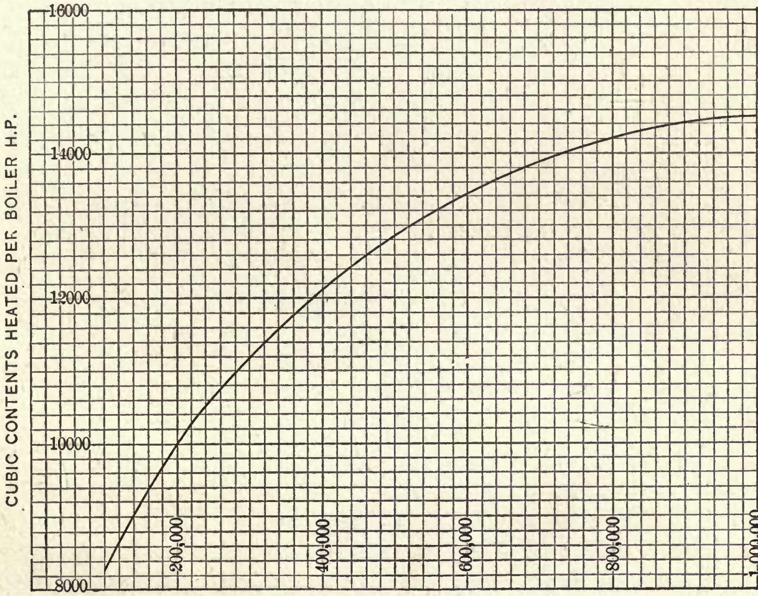

CHART 5.

Space Heated per Boiler Horse-power in Isolated Buildings under Conditions Stated.

Chart No. 5 shows by the curve the increased space that may be warmed per boiler horse-power in large buildings over that in smaller ones, since the former have less exposed surface per unit of contents.

Chart No. 6 is based on buildings ranging in size from 100,-000 to 1,000,000 cubic feet and in hight from 20 to about 60 feet,

according to the size. The buildings are assumed to be rectangular in plan, the length being twice the breadth in each case. The glass surface is assumed to be one-third of the total exposure; the equivalent glass surface of the roof is taken as one-tenth the total area of same. An allowance for reheating the buildings was made equivalent to an amount of heat that would raise a volume of air equal to the contents 20 degrees in one hour. This amount of heat would not actually raise the temperature of the air in the building that amount in the time stated, since the walls and machines or what not in the rooms must have their temperature raised as well as that of the air, and would absorb a large portion of the heat. The greater the amount of material in the rooms the less will be the fluctuation in temperature with intermittent heating, since the machinery or goods that become thoroughly warmed during the day, when surrounded by air at, say, 60 to 70 degrees temperature, store up heat which is given off during the night or at times when steam is shut off.

For direct radiating systems the charts will be of service in checking roughly the boiler horse-power required. They apply only to buildings exposed on all sides under the conditions stated as to glass surface, exposure, etc. For other conditions due allowances must be made.

On the basis of, say, 85 square feet of radiating surface per boiler horse-power, mains and risers to be computed as radiating surface unless covered, the horse-power indicated in Chart No. 6, multiplied by 85, gives roughly the square feet of radiating surface necessary for buildings of contents stated. For example: A building of 200,000 cubic feet requires 20 horse-power, per chart No. 6 $= 20 \times 85 = 1700$ square feet of radiating surface, a ratio of approximately 1 to 120 cubic feet. For 400,000 cubic feet, radiating surface $=$ about $33 \times 85 = 2805$, giving a ratio of 1 to 143 cubic feet, and so on.

Of course, the above is to be considered as only a rough approximation. The figure 85 is perhaps too conservative. For accurate work the wall and glass surface must be computed.

SIZE OF HEATERS WITH BLOWER SYSTEMS.

With the blower system the inleakage of cold air will be somewhat diminished by the pressure in the rooms maintained by the

fan. This pressure is scarcely measurable, however, and its effect in preventing inleakage of cold air will be neglected in this discussion. With air supply at 140 degrees and building at 70 degrees, half the heat supplied is carried away by the air escaping at 70 degrees, the other half being lost by transmission through walls, windows and roof. Under these conditions twice as much heat is necessary as with direct radiation.

If the frequent change of air incident to the blower system is necessary, or if ample exhaust steam is available, well and

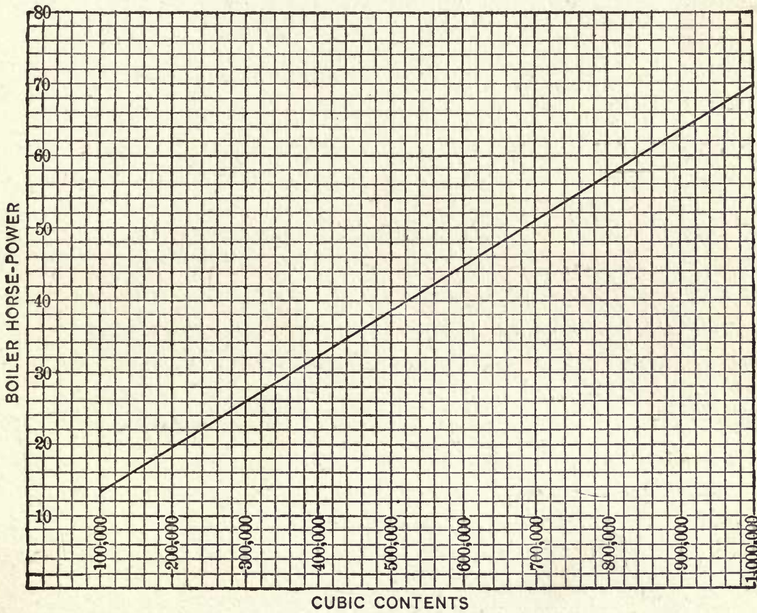

CHART 6.

Showing Approximate Boiler Horse-power Required to Heat Buildings having Various Cubical Contents.

good; otherwise the loss of heat over direct radiation is a serious one.

Since a building with the blower system, under the conditions stated, taking air from the outside, will require twice as much heat as with direct radiation, the boiler horse-power shown in Chart No. 6 must be doubled; for example, a building of 200,000 cubic feet will require about $2 \times 20 = 40$ horse-power, and on the basis

of 50 lineal feet of 1-inch pipe in the heater per horse-power, a not uncommon allowance, a 2000 lineal foot heater would be required.

With other conditions than 140 degrees, 70 degrees and zero, as stated above, greater or less boiler horse-power would be required with lower or higher inlet temperatures, respectively. A much higher inlet temperature than 140 degrees is not to be generally recommended. With the blower system the heater pipes, with low pressure steam and ordinary velocities of air between them, are generally rated to give out 2000 heat units or more per square foot an hour, or, say an average of 600 heat units per lineal foot of 1-inch pipe, corresponding to 55 lineal feet per horse-power.

RELATIVE LOSS OF HEAT FROM BUILDINGS HAVING THE SAME CUBIC CONTENTS.

The relative loss of heat from buildings having the same contents, but of different forms, is shown in the diagrams A B C and D of Fig. 27, each of approximately 125,000 cubic feet. Let each have glass equal to one-sixth the exposure, the equivalent glass surface of walls to equal the area of wall surface divided by 4, and let 1 square foot of roof be considered equivalent to one-tenth square foot of glass; the equivalent glass surface of each building is as stated under the different figures. Since the cubic contents is the same, the loss of heat would be roughly proportional to the equivalent glass surface in each. Long, low buildings require less horse-power per 1000 cubic feet than those more nearly cubical in form.

Building D, which is high in proportion to its floor area, would take considerably more horse-power per 1000 cubic feet than those represented by A, B or C.

The loss of heat by leakage of air would be greater in high buildings like D than in low ones like B and C, as they have a greater flue action involving greater leakage and have more wall surface in proportion to their contents than those shown in A, B and C.

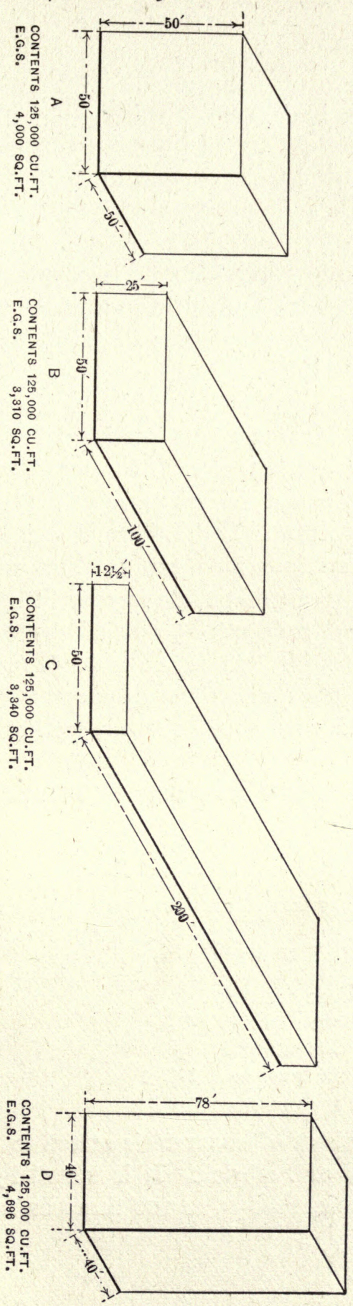

Fig. 27.—Showing Variation in Exposure in Buildings of Equal Cubic Contents.

TRANSMISSION OF HEAT THROUGH CONCRETE.

Tests made by Walter Kennedy on transmission of heat through concrete, published by the Armstrong Cork Co., Pittsburgh, showed for 4-inch concrete (1-3-5 mixture) a transmission of 25.5 and 26 B. T. U. per square foot per degree F. difference in temperature for 24 hours.

Figuring the loss per hour as is common in heating calculations, these values are equivalent to a trifle over 1.06 B. T. U. per hour per square foot per degree F. difference in temperature for concrete 4 inches thick.

From "Some Thermal Properties of Concrete," by Charles L. Norton, of the Massachusetts Institute of Technology, the following extracts are taken, with the permission of the National Association of Cement Users:

Professor Norton says the thermal conductivity is that property which determines how rapidly heat will travel through a substance. He states that there is very little data to be found as to this important property of any of the common materials of engineering and such data as are to be found are not at all concordant. As to the conductivity of concrete or its variations with temperature and with composition, practically nothing has been known.

Professor Norton's tests may be tabulated as follows:

TABLE XII.

THERMAL CONDUCTIVITY OF CONCRETE.

Temp. hot side of plate, degrees F.	Mixture.	Coefficient B. T. U. per 1° F. per square foot 1 inch thick per 24 hours.
95	Stone concrete 1·2·5	150
122	Stone concrete 1·2·4 Not tamped	76 to 114
95	Cinder concrete 1·2·4	56

The specific heat of concrete is stated to be slightly less than either red brick or firebrick. It is stated by Professor Norton that steel transmits heat 75 to 100 times as fast as the densest of stone concrete.

It is of interest to note the far greater conductivity of stone concrete than of cinder concrete, as given by Professor Norton. Kennedy's values for transmission through concrete 4 inches thick show a transmission for 70 degrees difference in temperature of about 75 B. T. U. per square foot per hour. This rate, it may be noted, is about twice as great as that for brick of the same thickness, the rate for the latter being that taken from a chart originally introduced into this country by the late A. R. Wolff and deduced from German values. These German values, by the way, are in pretty close agreement with those presented in this treatise, transformed into English units from the German by Professor Kinealy.

Table VII, page 43, shows a rate of heat transmission for stone walls to be about one and one-half times that for a solid brick wall of the same thickness.

In the absence of definite available data relating to the actual heat conductivity of concrete it is the author's opinion that an assumed rate of heat transmission through stone concrete of twice that through brick, as given in Table VI, page 43, is not very wide of the mark.

Doubtless actual tests will soon bring this matter of heat transmission down to a definite basis.

CHAPTER V.

HEATING EQUIVALENTS, SPECIFIC HEAT, HUMIDITY, THE HEATING AND COOLING OF AIR, ETC.

CONVERTIBILITY OF HEAT AND MECHANICAL ENERGY.

"The formal statement of the first law of thermodynamics is: Heat and mechanical energy are mutually convertible, and heat requires for its production and produces by its disappearance a definite number of units of work for each thermal unit. The mechanical equivalent of heat is designated by J."

THE MECHANICAL EQUIVALENT OF HEAT.

"The amount of mechanical work which is capable of generating one unit of heat is called the mechanical equivalent of heat. The mechanical or dynamical equivalent of heat is the number of units of energy or work to which one unit of heat is equivalent. The unit of work is the foot-pound, i.e., the amount of work required to raise one pound one foot. It has been determined experimentally that one heat unit (B. T. U.) is equivalent to 778 foot-pounds."

CONSERVATION OF ENERGY.

The following law, which is a statement of the doctrine of the conservation of energy, holds for all known forms of physical energy: "The total energy of any body or system of bodies is a quantity which can neither be increased nor diminished by any mutual action of these bodies, though it may be transformed into any of the forms of which energy is susceptible."—*Maxwell.*

HEAT PER HORSE-POWER.

Whenever mechanical work is done heat is given off. Thus, the heat due to the running of machines in a shop assists in the warming of the room. A horse-power is 33,000 foot-pounds per

minute. For each mechanical horse-power expended in whatever manner in factory, shop or elsewhere, $33{,}000 \div 778 = 42.4$ heat units are given off. A mechanical horse-power hour is equal, then, to 2545 heat units per hour, an amount equal to the loss of heat through over 30 square feet of glass, or that given off by 8 to 10 square feet of direct radiation.

LATENT HEAT OF VAPORIZATION.

"The latent heat of vaporization is the number of thermal units required to convert the unit mass (or weight) of a liquid at a given temperature into saturated vapor at the same temperature. At any pressure below the critical temperature, a substance may be converted wholly into saturated vapor by heating or cooling to the temperature of the boiling-point corresponding to that pressure. It is found experimentally that the heat of vaporization varies with the temperature at which the conversion from liquid into saturated vapor takes place. Hence the necessity for the introduction of the temperature restriction in the above definition."

The evaporation of water, a question which comes up in the consideration of air washers, involves an expenditure of heat.

The heat quantity expended in evaporation is very nearly

$$1092 \text{ B. T. U.} - 0.7\ (t^\circ - 32^\circ)$$

per pound of water evaporated at t° F.

Latent heat, commonly expressed in B.T.U., is the heat which disappears when a liquid is changed to a gas, as water to steam or water to aqueous vapor.

This heat again comes into evidence when the steam or vapor is condensed back to water, as when steam is condensed in a radiator, for example, or as when moisture is condensed from the atmosphere.

SPECIFIC HEAT AND THE HEATING AND COOLING OF AIR.

Different substances vary greatly in the amount of heat they must absorb to raise their temperature a given amount. The quantity of heat that must be imparted to a body to raise its tempera-

ture 1 degree in comparison with that required to raise an equal weight of water 1 degree is known as the "specific heat" of the body. Thus, the specific heat of air is 0.2375 (generally taken as 0.238)—that is, only about one-fourth as many heat units are required to raise 1 pound of air 1 degree as would be necessary to raise 1 pound water the same amount. The specific heat of water varies slightly, but this need not be taken into consideration except for scientific work.

To determine how many heat units are required to heat a given volume of air a stated number of degrees the quickest method is probably to multiply the volume in cubic feet by the degrees rise in temperature and divide the product by 55, this number representing approximately the number of cubic feet of air at 70 degrees that will be raised 1 degree by one heat unit. One cubic foot of dry air at 70 degrees temperature weighs 0.0747 pound, or 1 pound occupies 13.4 cubic feet. One heat unit will raise $\frac{1}{0.238}$ pound of air 1 degree, equal to 4.2 pounds of air 1 degree. Since 1 pound of dry air occupies 13.4 cubic feet, 1 heat unit will raise 4.2×13.4 cubic feet 1 degree $= 56$ cubic feet; 55 cubic feet is commonly used in making approximate calculations. On precisely the same basis it will be found that 1 heat unit will raise approximately 50 cubic feet of air at zero through 1 degree, zero air weighing 0.0864 pound to the cubic foot.

TABLE XIII.

THE WEIGHT OF MIXTURES OF AIR SATURATED WITH VAPOR PER CUBIC FOOT AT DIFFERENT TEMPERATURES.

Temperature.—F.	Weight in pounds of 1 cubic foot of dry air.	Temperature.—F.	Weight in pounds of 1 cubic foot of dry air.
0	0.086379	92	0.070717
12	0.084130	102	0.068897
22	0.082302	112	0.067046
32	0.080504	122	0.065042
42	0.078840	132	0.063039
52	0.077227	142	0.060873
62	0.075581	152	0.058416
72	0.073921	162	0.055715
82	0.072267		

See Table XVI, page 69, for weight of dry air per cubic foot at different temperatures.

COOLING AIR.

When the volume of air to be cooled is small, ice is generally used, each pound in melting absorbing about 142 heat units. Suppose, for example, it is desired to know the weight of ice that must be melted to cool 60,000 cubic feet of air per hour from 90 down to 80 degrees, the water from the melted ice to be discharged at 62 degrees temperature:

	Heat units.
1 pound of ice, in melting, absorbs	142
1 pound of water, when warmed from 32° to 62°, absorbs	30
Total heat units absorbed	172

The effect of condensing the moisture in the air must be allowed for, about 1000 heat units being given off per pound of moisture condensed.

One cubic foot of air at 90 degrees weighs 0.072 pound. Hence 60,000 cubic feet will weigh 4,320 pounds. Since the specific heat of air is 0.238, the number of heat units that must be absorbed by melting ice to cool this weight of air 10 degrees will be 4,320 pounds $\times$ 10 $\times$ 0.238 $=$ 10,250 heat units, approximately. Since 1 pound of ice melted and the water raised to 62 degrees absorbs 172 heat units, 10,250 $\div$ 172 heat units will be required, equal to about 60 pounds of ice per hour to cool 60,000 cubic feet of air 10 degrees F.

The ice would be most effective if it were crushed into small pieces so that the air would come in close contact with it. This, unfortunately, would seriously retard the flow of air, owing to the increased resistance over that when large cakes are used. With the latter arranged in properly constructed racks and provision made for retaining the water until its temperature has increased to within 20 or 30 degrees of that of the air, good results have been obtained; but practically one must expect the amount of ice required to exceed considerably the theoretical weight based on the volume of air cooled, since there are losses by transmission through surrounding partitions, walls, etc.

For large systems mechanical refrigeration should be used. It may be said in a general way that in small plants the consumption of 1 ton of coal is sufficient to produce 7 to 8 tons of commercial ice. The actual ice making capacity of a machine is only 50 to 60 per cent. of its so-called ice-melting capacity, which is expressed in tons capacity in 24 hours—that is, a 30-ton machine means a

refrigerating capacity in 24 hours equivalent to that produced by the melting of 30 tons of ice. The machine would produce, however, only 15 to 18 tons of real ice in the same period. For cooling air with a refrigerating plant, brine at, say, 8 to 12 degrees F. would be circulated by pumps through coils over which the air would be required to pass.

Unfortunately, the cooling of air does not make it agreeable. Its relative humidity is increased, which makes it less capable of absorbing moisture or perspiration from the body. Therefore the air should be dried by passing it over trays of calcium chloride, which has a great capacity for absorbing moisture, or it may be slightly heated after the chilling process to reduce its humidity.

It is unwise to cool the air in a room on a hot summer day more than about 10 degrees F. below the outdoor temperature, since to do so makes the room feel chilly to one entering from the outside.

When air is cooled down to the dew point, latent heat appears, due to condensation of moisture, and this is to be taken into consideration in cooling problems.

EVAPORATION AND HUMIDITY.

To moisten air water must be evaporated or steam must be injected into it. In either case about 1,000 heat units are necessary for the evaporation of 1 pound of water or the making of 1 pound of steam. Water evaporates very slowly when exposed in still air, the evaporation per square foot from a water surface in contact with still air at 70 degrees having a relative humidity of 40, being about 1-40 pound per hour. The rate of evaporation rapidly increases with an increase in temperature or the passage of air across the surface of the water. The capacity of air to absorb moisture increases rapidly with its rise in temperature—*e. g.,* air at 70 degrees can absorb about four times as much moisture as air at 30 degrees, as will be seen by referring to Table XIV:

TABLE XIV.

THE WEIGHT OF WATER VAPOR PER CUBIC FOOT OF SATURATED SPACE AT DIFFERENT TEMPERATURES.

Temperature.	Weight of vapor in grains per cubic foot.	Temperature.	Weight of vapor in grains per cubic foot.
0	0.54	50	4.09 = 4 approx.
10	0.84	60	5.76
15	0.99 = 1 approx.	70	7.99 = 8 approx.
20	1.30	80	10.95
30	1.97 = 2 approx.	90	14.81
40	2.88	100	19.79 = 20 approx.

1 pound avoirdupois = 7000 grains.
Approximately 1000 heat units are required to evaporate 1 pound of water.

The amount of heat and fuel necessary to moisten air is not generally appreciated. To illustrate this point take the amount of heat required to moisten air entering a furnace at 30 degrees, with a relative humidity of 65, so that a relative humidity of 50 will be maintained in the rooms kept at 70 degrees. Assume that 50,000 cubic feet of air per hour passes through the furnace: One cubic foot of saturated air at 30 degrees temperature contains, approximately, 2 grains of moisture, and with a relative humidity of 65 would contain 1.3 grains. Each cubic foot of air at 30 degrees expands to 1.08 cubic feet when heated to 70 degrees. One cubic foot of saturated air at 70 degrees contains about 8 grains of moisture. With 50 relative humidity 1 cubic foot of 70-degree air would contain 4 grains.

The amount of moisture that must be supplied by the evaporating pan in the furnace is the difference between 50,000 cubic feet per hour $\times$ 1.08 $\times$ 4 and 50,000 $\times$ 1.3. The difference equals 151,000 grains, or 21.6 pounds per hour.

Since about 1,000 heat units are required to evaporate 1 pound of water, 21,600 units per hour are absorbed, equal to the heat utilized from the burning of 2½ pounds of coal.

If an attempt is made by specially provided means to raise the relative humidity in the room to, say, 50, in cold winter weather, the moisture will condense on the windows and they will become frosted.

A relative humidity of about 30 is said to be about as high as one can secure without trouble from condensation on single windows in severe winter weather.

ACTUAL AND RELATIVE HUMIDITY.*

By actual humidity (A.H.) is meant the weight of water vapor in a given unit volume of space or air as the number of grains contained in a cubic foot of air.

By relative humidity (R.H.) is meant the ratio in hundredths between the quantity present in that volume of space or air and the quantity it would contain if saturated.

Actual humidity remaining constant, relative humidity is determined by temperature.

Dew Point.—When the humidity of any space is raised to saturation it may be said to have reached the saturation point, as any excess of vapor must be precipitated in fog or dew.

When the temperature of any given space is lowered to such a point that the contained vapor saturates the space, that saturating temperature is called its dew point.

It is pointed out in pamphlet W. B. 235, treating of humidity, etc., by C. F. Marvin, and published by the U. S. Department of Agriculture, that a false notion is widely prevalent that air has a certain capacity for moisture. It is pointed out that the presence of moisture in a given space is independent of the presence or absence of air in the same space except that the air retards the diffusion of the vapor particles. It is more correct to say that the *space* is partly saturated with moisture.

COOLING AND CONDITIONING AIR.†

"The cooling of 10,000 cubic feet of air a minute, or 600,000 cubic feet an hour, from 95 to 80 degrees would require an abstraction of heat from it which would melt 1,150 pounds of ice. But air so cooled would be more uncomfortable and dangerous than the hotter air of 95 degrees because of its excessive humidity. To remove the excess of moisture by cooling would require an additional extraction of heat, varying with the temperature and relative humidity of the outside air, equal to the

* From S. H. Woodbridge's Technology Notes.

† Extract from Professor Woodbridge's Report on the Ventilation of the U. S. Senate Chamber, dated December 14, 1895.

melting of from 600 to 1,000 pounds of ice, or a total refrigerating effect of about 1 ton of ice an hour.

Either of two methods may be followed for making the treated air salubrious and agreeable. The whole quantity of air cooled may be brought down to so low a temperature as to precipitate the necessary moisture for drying it, and then warmed again by artificial heating to the temperature and dryness essential to comfort; or a part only of the air may be so sharply chilled as to remove the weight of moisture necessary to insure dryness, and this chilled and dried air may then be passed on and mixed with the untreated part, resulting in the drying and cooling of the entire volume of air. . . .

By means of modern mechanical refrigerating appliances from 15 to 20 tons of refrigerating effect may be realized from 1 ton of coal, and such cooling and drying as is above mentioned could, therefore, under the best working conditions, be had by a fuel expenditure of from 150 to 200 pounds of coal an hour."

COST OF INCREASING HUMIDITY.

The statement is sometimes met with that air may be humidified without extra cost, that is, that a saving may be made in the loss of heat by transmission through walls and glass sufficient to offset the cost of adding moisture to the air. The saving in heat by transmission is due to the lower room temperature that may be maintained with a relatively higher humidity, than the temperature that would be necessary to produce practically the same degree of comfort when the air in the room is of the low relative humidity common in heated buildings during the winter.

This statement may easily be proven to be false, a much larger amount of heat being used up in evaporating the water to secure the increased humidity than is saved by the lessened transmission through walls and glass, due to a lower room temperature.

EXPANSION OF AIR AND ABSOLUTE TEMPERATURE.

Air expands and contracts with changes in temperature according to a known law—viz., for each degree rise or fall in temperature from 32 degrees F. air expands or contracts $^1/_{491}$

of its volume at that temperature. If a cubic foot of air be heated through 491 degrees from 32 degrees, or to 523 degrees, it will double in volume. On the other hand, if a cubic foot of air be cooled through 491 degrees from 32 degrees, or to 459 degrees below zero, it will theoretically contract $\frac{491}{491}$ of its original bulk, or will entirely disappear. This point, 459 degrees below zero, or more accurately 459.4 degrees, is known as absolute zero, and is the point from which the expansion of air is reckoned in determining its relative volume at different temperatures, the volume being proportional to the absolute temperature. For convenience in making ordinary calculations 460 degrees F. below zero may, with sufficient accuracy, be considered absolute zero. Hence the absolute temperature of a body is equivalent to 460 degrees plus its Fahrenheit temperature. Suppose, for example, we wish to determine how much space 1 cubic foot of air entering a furnace at 0 degree F. will occupy when heated to 140 degrees F. Since the volume varies in proportion to the absolute temperature, we have:

Absolute temperature of air at $0°F = 0° + 460° = 460$
Absolute temperature of air at $140°F = 140° + 460° = 600$
} Volume at 0° is to volume at 140° as 460 is to 600.

Hence, volume at 140 degrees $= \frac{600}{460} \times$ volume at 0 degree; volume at 140 degrees = 1.3 cubic feet.

TABLE XV.

THE APPROXIMATE VOLUME TO WHICH 1 CUBIC FOOT OF AIR AT 0° WILL EXPAND WHEN HEATED TO THE TEMPERATURES STATED IN THE TABLE. VOLUME OF AIR AT 0° = 1 CUBIC FOOT.

Volume when heated to— Degrees.	Cubic feet.	Volume when heated to— Degrees.	Cubic feet.
10	= 1.02	110	= 1.24
20	= 1.04	120	= 1.26
30	= 1.06	130	= 1.28
40	= 1.09	140	= 1.30
50	= 1.10	150	= 1.33
60	= 1.13	200	= 1.44
70	= 1.15	300	= 1.65
80	= 1.17	400	= 1.87
90	= 1.20	500	= 2.09
100	= 1.22		

TABLE XVI.

THE WEIGHT OF DRY AIR PER CUBIC FOOT AT DIFFERENT TEMPERATURES.

Temperature. Degrees F.	Weight of a cubic foot in pounds.	Temperature. Degrees F.	Weight of a cubic foot in pounds.
0	0.0864	112	0.0694
12	0.0842	122	0.0682
22	0.0824	132	0.0671
32	0.0807	142	0.0660
42	0.0791	152	0.0649
52	0.0776	162	0.0638
62	0.0761	172	0.0628
72	0.0747	182	0.0618
82	0.0733	192	0.0609
92	0.0720	202	0.0600
102	0.0707	212	0.0591

VELOCITY OF AIR IN FLUES.

The velocity of air in a flue is governed by its hight and the difference between the inside and outside temperature. Suppose we have a flue 1 square foot in area and of hight h, represented in Fig. 28.

The air in the flue is balanced by a column of colder outside air of hight H, leaving an unbalanced force represented by the hight $(h - H)$, tending to produce a velocity at the base of the flue equivalent to that developed by a body falling freely through a distance represented by the hight $(h - H)$.

The velocity acquired by such a body, neglecting friction, is expressed by the equation

$$v = \sqrt{2gh} \quad \ldots\ldots\ldots\ldots (a)$$

Here v = velocity in feet per second, g = the acceleration in feet per second due to gravity, = 32.2 feet, h = the hight through which the body falls—in this case represented by $(h - H)$.

Now let

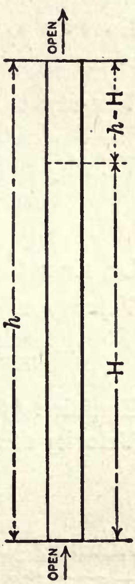

Fig. 28.—Flue Diagram.

w_0 = the weight per cubic foot of outside air.
w_F = the weight per cubic foot of air in the flue.
t_0 = the absolute temperature of the outside air = Fahrenheit temperature + 459.4°.
t_F = the absolute temperature of the air in the flue = Fahrenheit temperature + 459.4°.

We have seen that the velocity at which the air enters the base of the flue is expressed by

$$v=\sqrt{2\ g\ (h-H)} \ldots\ldots\ldots\ldots (b)$$

Now since the columns of air represented by h and H balance each other we have weight of column h = weight of column H; *or,*

$$h\ w_F=H\ w_O \ldots\ldots (c) \text{ hence } H=\frac{h\ w_F}{w_O} \ldots\ldots (d)$$

The density of the air, or its weight per cubic foot, varies inversely as the absolute temperature; hence we may substitute for $\frac{w_F}{w_O}$, $\frac{T_O}{T_F}$ equation (d) becoming $H=h\ \frac{T_O}{T_F} \ldots\ldots (e)$

Substituting this value of H in (b) we have

$$v=\sqrt{2\ g\left(h-h\frac{T_O}{T_F}\right)}=\sqrt{2\ g\ h\ \left(\frac{T_F-T_O}{T_F}\right)} \ldots\ldots (f)$$

Now the weight of air leaving the flue must be equal to the weight of air entering—that is,

Velocity of air leaving flue $\times\ w_F$ = velocity of air entering flue $\times\ w_O \ldots\ldots (g)$

Velocity of air leaving flue =

$$\frac{\text{velocity of air entering flue} \times w_O}{w_F} \ldots\ldots (h)$$

Or, since the weight varies inversely as the absolute temperature,

Velocity of air leaving flue =

$$\frac{\text{velocity of air entering flue} \times T}{T_O} \ldots\ldots (i)$$

Equation (f) gives the velocity of the air entering the flue, hence Velocity of air leaving or passing through the flue =

$$\frac{T_F}{T_O}\sqrt{2\ g\ h\ \left(\frac{T_F-T_O}{T_F}\right)}$$

Allowing 50 per cent. for friction, and substituting the value of g = 32.2, the velocity in feet per minute in the flue is

$$V=240\ \frac{T_F}{T_O}\sqrt{h\ \left(\frac{T_F-T_O}{T_F}\right)}$$

from which the following table is calculated:

TABLE XVII.

THE APPROXIMATE VELOCITY OF AIR IN FLUES OF VARIOUS HIGHTS.

Outside temperature 32 degrees. Allowance for friction 50 per cent. in flue one square foot in area.

Hight of flue.	Excess of temperature of air in the flue over that outdoors.											
	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°	120°	140°
Feet.	Velocity of air in feet per minute.											
5	77	111	136	159	179	199	216	234	250	266	296	325
10	109	156	192	226	254	281	306	330	354	376	418	460
15	133	192	236	275	312	344	376	405	432	461	513	565
20	154	221	273	319	359	398	434	467	500	532	592	650
25	173	248	305	357	402	445	485	522	560	595	660	728
30	189	271	334	390	440	487	530	572	612	652	725	798
35	204	293	360	423	475	527	574	620	662	705	783	862
40	218	311	386	452	508	562	612	662	707	753	836	920
45	231	332	408	478	538	597	650	700	750	800	887	977
50	244	350	432	503	568	630	685	740	790	843	935	1030
60	267	383	473	552	622	690	750	810	865	923	1023	1125
70	289	413	510	596	671	746	810	875	935	995	1105	1215
80	308	443	545	638	717	795	867	935	1000	1065	1182	1300
90	327	470	578	678	762	845	920	990	1060	1130	1252	1380
100	345	495	610	713	802	890	970	1045	1118	1190	1323	1455

Since the volume of air in cubic feet per minute discharged by a flue equals the velocity in feet per minute multiplied by the area in square feet,

$$\text{Velocity} = \frac{\text{volume}}{\text{area}} \qquad \text{Area} = \frac{\text{volume}}{\text{velocity}}$$

Example: Find the area of a flue 20 feet high that will discharge 3000 cubic feet per minute, when the excess of temperature in the flue over that outdoors is 40 degrees.

Opposite 20 in left-hand column and under 40 on upper line is the number 319, representing the velocity in feet per minute. The volume $3000 \div 319 = 9.4$ square feet, the required area. In estimating the effective hight of a warm-air flue from a furnace, consider the flue to begin 2 feet above the grate.

CHAPTER VI.

HEATING WATER.

The question frequently comes up how to determine the heating surface required to heat a given volume of water a certain number of degrees in hot water storage tanks or generators, as shown in Fig. 29. The proportions of feed water heaters in connection with boilers give a basis for such calculations, these heaters of the closed tubular type having 1-3 to ½ square foot of heating surface per boiler horse-power.

HEATING WATER BY SUBMERGED STEAM PIPES.

Taking the greater amount as a basis, ½ square foot of heating surface is expected to heat about 30 pounds of water per hour from, say, 50 to 200 degrees, that is $30 \times 150 = 4500$ heat units. In other words, a square foot is rated to transmit 9000 heat units per hour.

Suppose the exhaust steam pressure is 2 pounds, corresponding to a temperature of about 220 degrees, the average water temperature is $(200 + 50) \div 2 = 125$ degrees, making the average difference in temperature between the steam and the water $220 - 125 = 95$ degrees. Hence the number of heat units transmitted per square foot of heating surface per hour per degree difference in temperature is $9000 \div 95$, or about 100 heat units in round numbers.

Low pressure steam coils surrounded by air at 70 degrees give off only about 2 heat units per degree difference in temperature per hour, whereas when immersed in water they condense steam per degree difference about 50 times as rapidly—a striking fact.

To take a practical example, suppose it is desired to compute the heating surface in brass pipe required to raise the temperature of the water in a 4000-gallon tank from 70 degrees to 160 degrees in two and one-half hours with steam at 5 pounds pressure. The given number of gallons is equivalent to 4000×8 1-3 (number of pounds per gallon) = 33,333 pounds. The increase

in temperature is 90 degrees. Total number of heat units required is therefore 33,333 × 90 = 2,999,970. The number of heat units required per hour is thus approximately 1,200,000. The average difference in temperature between the steam and water is 228 — 115 = 113 degrees. Since 1 square foot of heating surface with 1 degree difference between the temperature of the steam and water gives off approximately 100 heat units per hour, with 113 degrees difference approximately 11,300 heat units will be given off in an hour, and 1,200,000 ÷ 11,300 = 106 square

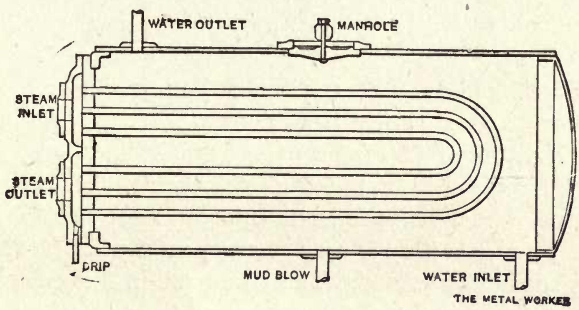

Fig. 29.—Hot Water Storage Tank Heated by Steam.

feet, the area of the heating surface required, which is 1 square foot to approximately 37 gallons capacity.

HOT WATER GENERATORS.

Hot water generators, so-called, otherwise known as coil boilers or hot water storage tanks, commonly have about 1 linear foot of 1-inch pipe to each 5 gallons capacity—that is, about 1 square foot of heating surface to each 15 gallons capacity. Such boilers are commonly assumed to be capable of heating their contents at least once an hour from, say, 60 to 160 degrees.

To heat 300 gallons per hour, for example, 100 degrees would require the expenditure of 300 × 8 1-3 × 100 = 250,000 heat units (8 1-3 representing the approximate weight of 1 gallon of water in pounds). With a coil based on the proportions stated, 1 square foot to 15 gallons capacity, the heating surface is 20 square feet and the heat emitted per square foot per hour would be 250,000 ÷ 20 = 12,500 heat units.

With steam at 228 degrees and average water temperature at

110 degrees the difference is 118 degrees; the transmission per square foot per hour per degree difference is 12,500 ÷ 118 = 106 heat units.

A hot water generator of even moderate size when heating the contents once an hour condenses an immense amount of steam. Take, for example, one of, say, 300 gallons capacity. To heat this volume from, say, 50 to 160 degrees requires 300 × 8 1-3 × 110 = 275,000 heat units. The condensation of 1 pound of steam at 5 pounds pressure gives off 954.6 heat units; therefore nearly 300 pounds of steam would have to be condensed in an hour, equivalent to about 10 boiler horse-power, or the consumption of 35 to 40 pounds of coal.

In office buildings and apartment houses at certain periods the volume of water drawn from the hot water generator is equal to a per hour rate many times in excess of the average per hour requirements throughout the day. The generator, or hot water storage tank, must be made large enough to meet these demands, just as a storage battery is used to carry an electric plant through certain periods of overload. The steam coil in the generator then has several hours in which to make good the sudden large drafts that occur at intervals.

BOILING LIQUIDS IN VATS.

It is a well-known fact that when water is heated in an open vessel to the boiling point, 212 degrees F., its temperature cannot be increased. If more heat is applied it simply causes the water to boil more rapidly. The amount of heat required to evaporate 1 pound of water at a temperature of 212 degrees into steam at the same temperature is, neglecting decimals, 966 heat units. This is known as the latent heat. The same number of heat units are given up by the steam when it is condensed back into water. For example, an ordinary heating coil condensing about 1-3 pound of steam per square foot per hour gives off a little more than 300 heat units, or about one-third of the latent heat in a pound of steam.

In computing the amount of coil necessary to evaporate a given amount of water in a stated time proper allowance must be made for the latent heat necessary to evaporate the water after

sufficient heat has been applied to bring it to the boiling point. Since the heat given off by the coil depends on the difference in temperature between the steam inside and the water outside one should have 20 to 40 pounds steam pressure in order to provide a reasonable excess of temperature in the steam over the water. For boiling thick, heavy liquids considerably more heating surface is necessary than for boiling water, on account of the more sluggish circulation. The difference in the specific heats also enters in.

HEATING SMALL SWIMMING POOLS.

Hot water generators fitted with steam coils, as shown in Fig. 29, are sometimes used to heat small swimming pools, the water being admitted to the latter through concealed pipes placed near the bottom.

When connected with a gravity return system of steam heating no more attention is necessary with regard to maintaining the proper amount of water in the steam boiler than if the steam coil in the hot water generator were a large radiator, since the condensation returns to the boiler, provided the generator is located well above the water line.

To compute the size of coil required with this method of heating take, for example, a pool 12 x 30 feet in plan and 5 feet in average depth. Its contents, 1,800 cubic feet, multiplied by 62½, the number of pounds 1 cubic foot of water weighs, gives about 112,500 pounds to be heated.

Suppose the water to be continuously changing at the rate of one complete change every ten hours, equivalent to 11,250 pounds of water per hour. If the water in the street mains is at, say, 50 degrees, and that in the pool 75 degrees, $11,250 \times 25 = 281,250$ heat units must be supplied per hour.

It has been pointed out that 1 square foot of heating surface in the generator will give out about 12,500 heat units per hour; therefore $281,250 \div 12,500 = 22.5$, or about 22 square feet of coil would be necessary when using steam of, say, 5 pounds pressure. On the customary basis of 1 square foot of heating surface to 15 gallons capacity, 22 square feet of surface would correspond to a 330-gallon boiler, which, from experience, the writer has found gives good service under the conditions stated.

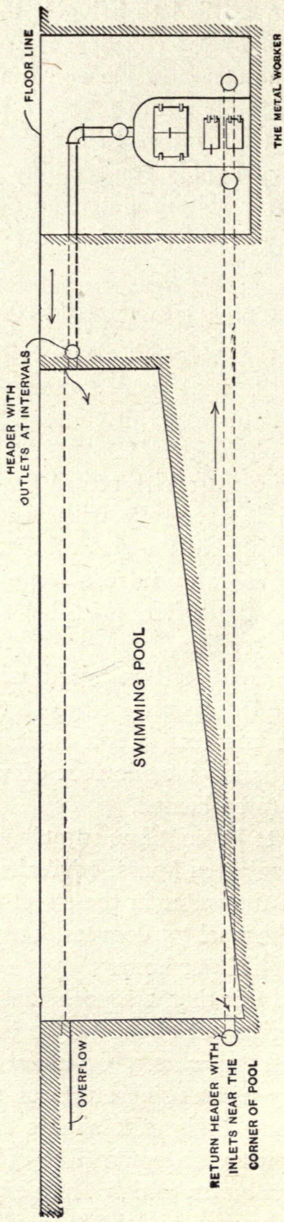

Fig. 30.—Arrangement of Heater Connected with Swimming Pool.

HEATING LARGE SWIMMING POOLS.

For heating large pools one of three methods is commonly used:

1. Steam is admitted directly to the water in the pool through one of the devices on the market for muffling the sound.

2. Steam coils are submerged in the pool.

3. The water is made to circulate through a boiler or heater, as shown in Fig. 30, the pool being practically a huge expansion tank.

AMOUNT OF STEAM AND SIZE BOILER REQUIRED.

The amount of steam that must be admitted to heat the water in a pool will depend on the volume, the temperature and the time in which the heating must be done.

Take, for example, a pool 20 x 80 feet, with an average depth of 7 feet, equal to 11,200 cubic feet, in which the water is to be heated from the street temperature of, say, 50 degrees, to a temperature of 80 degrees during a period of ten hours. Water at the street temperature weighs approximately 62½ pounds per cubic foot; therefore 11,200 cubic feet of water to be raised 30 degrees in ten hours will require a number of heat units per hour equal to

$$11{,}200 \times 62.5 \times 30 \div 10 = 2{,}100{,}000 \text{ heat units.}$$

Suppose the steam be admitted at low pressure, say 5 pounds. One pound at that pressure will supply 955 heat units when condensed, and the water, in cooling from 228 degrees, the temperature of the steam at 5 pounds pressure, to 80 degrees, will give up 148 heat units more, making a total of 1,103 heat units per pound. This figure is contained in the total number of heat units required about 1,903 times—that it, 1,903 pounds of steam must be condensed in one hour.

The boiler capacity required is equal to 2,100,000 ÷ 33,305 (which is a boiler horse-power expressed in heat units) = 63 horse-power. The above makes no allowance for the loss of heat by evaporation—a subject previously discussed—nor for losses through the walls or the bottom of the tank.

AMOUNT OF STEAM PIPE REQUIRED.

To ascertain the amount of steam pipe required with steam at, say, 5 pounds pressure, the pipes to be placed around the tank in recesses near the bottom, other conditions to be as stated above, proceed as follows: The average difference in temperature between the steam and water is

$$228 - \left(\frac{50 + 80}{2}\right) = 228 - 65 = 163 \text{ degrees.}$$

The discussion of feed water heaters showed that it is approximately correct to reckon on 100 heat units being given off per hour by the steam to the surrounding water per degree difference in temperature. Hence, with 163 degrees difference, we should expect to transmit to the water 16,300 heat units for each square foot of brass pipe installed. If galvanized wrought iron pipes are used we should expect to get only about 70 per cent. of the heat stated above, or 11,410 heat units per square foot per hour.

The total heat units—viz., 2,100,000—divided by 11,410, gives 184 square feet, or about 368 linear feet, of 1½-inch pipe that would be required to meet the conditions stated. If the water were to be heated in a shorter time proportionately more surface would be required.

The above computations have as a basis the heat given off by the pipes or tubes in feed water heaters where the circulation of water is comparatively rapid. With coils submerged in tanks the movement of water over them is sluggish and the heat is taken up from the pipes less rapidly, hence it is wise to add 25 to 50 per cent. to the computed amount of pipe according to its location to allow for this sluggish circulation.

SIZE BOILER REQUIRED.

If a boiler is to be used, as in the third method mentioned, the amount of coal to be burned, the size of the grate and the size of the boiler may be determined as follows:

The total number of heat units required per hour, as computed above, is 2,100,000. Assuming that 1 pound of coal will give up in this case 9,000 heat units, since the boiler will be more efficient than usual, due to the cooler heating surfaces, the coal required will be $2{,}100{,}000 \div 9{,}000 = 233$ pounds per hour. With a regu-

lar attendant at least an 8-pound rate of combustion per square foot of grate surface per hour may be safely assumed. The grate surface therefore should be $233 \div 8 =$ approximately 30 square feet.

The water in the boiler being only 80 odd degrees instead of, say, 228 degrees, with 5 pounds pressure, less heating surface is required, in proportion to the grate area than with ordinary heating boilers to give the same efficiency. Assuming, as a rough approximation, that the average temperature of the gases in the boiler or heater is 700 degrees, the effectiveness of the heating surface in the two cases would be in the proportion of $\frac{700 - 80}{700 - 228} = \frac{620}{472}$; that is, only about $472 \div 620 = 76$ per cent. as much heating surface per square foot of grate would be required in the boiler used for heating water to 80 or 90 degrees as would be needed in ordinary low pressure boilers.

The gist of this is that a heater for the purpose stated could have an abnormally large grate in proportion to its size and still be economical in the use of coal.

TANK HEATERS.

Tank heaters are commonly rated by manufacturers to heat one-half to three-fourths as many gallons of water per hour as the number of square feet of direct radiating surface they will supply; that is, a heater with a grate 20 x 24 inches connected as shown in Fig. 31, would be rated to carry, say, 600 square feet of radiating surface, or to heat 300 to 450 gallons of water per hour. Manufacturers fail to give the temperatures from and to which the water is heated, but for apartment houses and the like the tank temperature should be kept as a rule at about 160 degrees. Therefore the water must be heated on an average at least 100 degrees above that of the city supply.

It is a simple matter to show on a heat unit basis that a much greater expenditure of heat is necessary to raise 300 gallons—that is, 2,500 pounds—of water 100 degrees than to supply 600 square feet of radiating surface, the heat losses being 250,000 and 90,000 respectively. Therefore tank heaters are commonly overrated. This fact, however, seldom becomes apparent, as the

large capacity of the storage tank enables the heater to heat the water at night, or when little water is drawn, so that time and storage capacity help out the overrated heater.

If one knows approximately the number of gallons of water that must be heated per hour to a given temperature from that of the city supply, the size heater may be readily determined on the heat unit basis. For small heaters, having, say, not over 2 square feet grate surface, the rate of combustion should not exceed 3 pounds per square foot of grate per hour. Larger heaters may

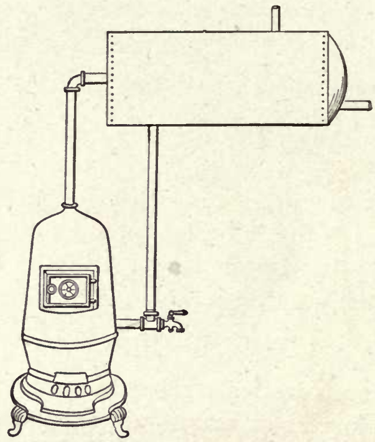

Fig. 31.—Tank Heater Connections.

be rated to burn 4 to 5 pounds or even more with frequent attention.

Example: What size will be required to heat 75 gallons per hour 90 degrees? The product: 75 (gallons) $\times$ 8 1-3 (number of pounds of water in one gallon) $\times$ 90 (the temperature range), gives the number of heat units involved. Dividing this product by 3 (number of pounds of coal burned per square foot of the grate per hour) $\times$ 8000 (the number of heat units utilized per pound of coal) gives 2.3 as the number of square feet of grate surface required. The above basis of computation will be found convenient in determining the size heater to be used for a baptistry pool, when the volume to be heated, the time and the temperature to be attained are known.

By installing a storage tank of good size a small heater may be made to do as good service as a much larger one with a small tank. That is, with a large tank, holding several times the probable maximum hourly volume required, a sudden draft in excess of the ability of the heater to make good immediately is not accompanied by a lowering of temperature at the faucets, as would be the case with a small tank. The assumption is sometimes made that, unless stated to the contrary, heaters rated to supply tanks of certain capacities are capable of heating a volume of water equal to the tank capacity in one hour. As just stated, it is better that the tank capacity should be several times the average hourly consumption.

Taking the ratings of a prominent manufacturer:

Heater with 12-inch grate is rated to supply a 200-gallon tank.
Heater with 15-inch grate is rated to supply a 325-gallon tank.
Heater with 18-inch grate is rated to supply a 485-gallon tank.

Averaging these gives 1 square foot of grate to 266 gallons tank capacity. Even with a rapid rate of combustion, say 5 pounds per square foot per hour, a square foot of grate would heat only about 48 gallons per hour 100 degrees, showing the tank capacity stated in these ratings to be over five times the hourly heating capacity of the heaters.

Suppose the water is heated from a street main at a temperature of 60 degrees to only 120 degrees; then 1 square foot of grate with a 5-pound rate of combustion would heat 80 gallons per hour, or only about one-third the rated tank capacity per square foot stated in the manufacturer's ratings. On the basis of 80 gallons per hour heated from 60 to 120 degrees per square foot of grate, a 320-gallon boiler, contents to be used once an hour, should have a heater with at least 4 square feet of grate surface, equivalent to a grate 27 inches in diameter. This would be uncommonly large for a tank of the size stated, showing that with the ordinary proportions of grate to tank capacity it must not be expected that the contents of the tank will be heated in less than several hours.

WATER BACKS AND GAS HEATERS.

Water backs in ranges ordinarily have 2 to 2½ square inches of heating surface per gallon capacity in the hot water boilers

with which they are connected. The ordinary temperature of water from city mains would be 50 degrees or more, running up to 70 degrees or so in summer. While 160 degrees is a common temperature for the hot water supply in large buildings having a separate heater, the temperature of a domestic supply is generally

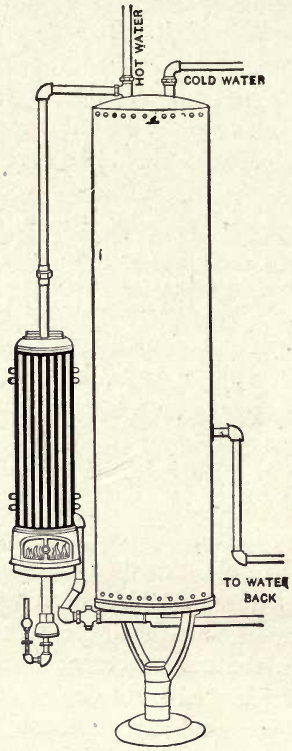

Fig. 32.—Gas Heater Connected with Range Boiler.

much lower, say not over 130 degrees as a rule, though in some cases much higher—even above boiling temperature at atmospheric pressure. Now, under the most favorable conditions the hot water supply must be heated from 70 to 130 degrees, equal to 60 degrees rise.

Take, for example, a 40-gallon boiler connected with a water back of 100 square inches area. To heat 40 gallons per hour 60 degrees would take $40 \times 8\ 1\text{-}3 \times 60 = 20{,}000$ heat units, which

with a water back area of about 2-3 square foot would mean over 30,000 heat units per square foot per hour transmitted to the water. Such a rating would be altogether too high with the proportions of water back and tank capacity just stated.

Similar surfaces in furnaces with combination heaters are seldom rated to carry over 75 square feet of direct radiating surface for each square foot of heating surface exposed to the fire; this is equivalent to only 75 × 150 (150 being the heat units given off per square foot of radiating surface per hour) =11,250 heat units. This is only a little more than the heat given off per square foot by steam coils in contact with water. The low rating is due to the chilling effect of the coil or water back on the fire in contact with it.

For ordinary working conditions the writer believes a rating of 10,000 heat units per square foot per hour for water backs to be a fair one, but with a brisk fire, as on ironing days, the water back will probably take up as much as 15,000 heat units per square foot per hour.

It is pretty difficult to determine in advance in any household the approximate volume of hot water that must be supplied. Families of the same size differ greatly in the amount of water they are in the habit of using. A water back to meet maximum use would be altogether too large for ordinary use. The best way to meet excessive occasional demands is to use a gas heater, connected as shown in Fig. 32, in addition to the water back.

Tests of ordinary gas heaters used in connection with hot water boilers are stated to have shown efficiencies of 52 to 74 per cent., when burning gas having a heating power of 540 heat units per cubic foot.

COILS FOR HEATING WATER.

Coils for heating the domestic water supply are frequently placed in hot water or steam heaters. On the basis of 15,000 heat units per square foot per hour; to heat 40 gallons per hour, for example, from, say, 60 to 130 degrees, or through 70 degrees, 40 × 8 1-3 × 70 = 23,310 heat units would be necessary, requiring about 1½ square feet of heating surface, equal to 4½ lineal feet of 1-inch pipe or 3.5 feet of 1¼-inch pipe.

If the coil is suspended in the combustion chamber above the fire a much lower rating must be assumed. It is well to arrange

the coil so that the fire may be brought in contact with it when desired by carrying a deep bed of coal on the grate. The heating capacity of a coil placed above the fire varies greatly with the condition of the fire on top; a bright fire giving good results and one black on top heating the water but little. As a rule it is a rather unsatisfactory way to heat a water supply unless the fire is run very regularly. A good sized tank should be used to avoid overheating.

An independent hot water stove or tank heater is generally to be preferred. A rating as high as 15,000 heat units should hardly be used, except when the fire is sure to receive careful attention. A rating of 10,000 to 12,000 heat units would represent more closely what could be expected in ordinary practice.

CHAPTER VII.

CAPACITIES OF PIPES FOR HOT WATER HEATING.

THE FLOW OF WATER IN PIPES.

The force causing circulation in a hot water heating system, due to the difference in temperature of the water in the supply and return pipes, is very slight and amounts to only 1 grain, or 1-7000 pound per square inch per degree difference in tempera-

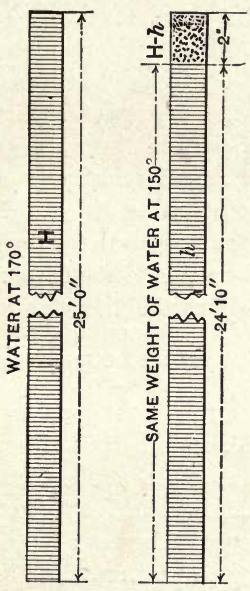

Fig. 33.—Head of Water Causing Flow.

ture per foot of hight. In ordinary two-pipe up-feed systems the hight is to be considered as that between the middle of the boiler and that of the topmost radiator. Suppose the supply and return risers to be 25 feet high with 20 degrees difference in temperature, then the excess of weight in the return over that in the supply line will be $25 \times 20 = 500$ grains $=$ 1-14 pound for each square

inch cross sectional area. Since 1 pound pressure is equivalent to about 2.3 feet, 1-14 pound is equal to a head of, approximately, 0.165 feet, or about 2 inches.

Put in another way, let H in the accompanying sketch represent the hight of a column of water at 170 degrees and *h* the hight of a column of equivalent weight at 150 degrees. Let H = 25 feet, then

$$h = \frac{25 \times 60.801 \text{ (weight of 1 cubic foot at 170 degrees)}}{61.204 \text{ (weight of 1 cubic foot at 150 degrees)}}$$

= 24.835 feet. Then, H — *h*, the hight representing the head or unbalanced force causing circulation of the water, is 0.165 foot, or about 2 inches, as above.

Were it not for friction the velocity corresponding to this head would be about 195 feet per minute, since the velocity in feet per second, neglecting friction, is approximately eight times the square root of the head, expressed in feet. Friction, however, plays a very important part in the laws governing the flow of water in pipes, and the actual velocity is only a fraction of the theoretical velocity, computed as above. The resistance to the flow is proportional to the length of the pipe to the square of the velocity, and decreases as the diameter increases. That is, the resistance varies inversely as the diameters.

VOLUME OF WATER TO SUPPLY RADIATORS.

The volume of water that must pass through a radiator of a given size to maintain a certain output of heat may be determined as follows: Take, for example, a direct radiator of 100 square feet, in which the water is cooled 15 degrees and which gives off 150 heat units per square foot per hour. The heat given off equals 100 × 150, or 15,000 heat units per hour. Since the water is cooled 15 degrees, each pound gives up 15 heat units; therefore, 1000 pounds must be cooled in an hour. Suppose the water enters at 170 degrees. Table XVIII, herewith, shows that water at this temperature weighs 60.801 pounds per cubic foot. Therefore, 1000 ÷ 60.801, or 16.41 cubic feet, must pass through a 100 square foot radiator to give up the heat units stated. This number of cubic feet multiplied by 7½ gives the number of gallons required —viz., 123.1.

TABLE XVIII.

VOLUME AND WEIGHT OF DISTILLED WATER.

"Weisbach."

Temperature in degrees F.	Weight of a cubic foot in pounds.	Temperature in degrees F.	Weight of a cubic foot in pounds.
32	62.417	170	60.801
39.1	62.425	180	60.587
40	62.423	190	60.366
50	62.409	200	60.136
60	62.367	210	59.894
70	62.302	212	59.707
80	62.218	220	59.641
90	62.119	230	59.372
100	62	240	59.096
110	61.867	250	58.812
120	61.720	260	58.517
130	61.556	270	58.214
140	61.388	280	57.903
150	61.204	290	57.585
160	61.007	300	57.259

THE VELOCITY IN HOT WATER HEATING PIPES.

To compute the velocity in pipes, suppose, for example, a 2-inch pipe supplies 200 square feet of surface, the water to drop 20 degrees in passing through the radiator. This amount of surface will give off about 200×150 heat units $= 30{,}000$ heat units per hour. Suppose the water enters at 170 degrees, weighing 60.801 pounds per cubic foot. Each pound gives up 20 heat units; then, $30{,}000 \div (60.801 \times 20)$ 24.6 cubic feet must pass through the radiator per hour, equal to about 0.41 cubic foot per minute. A 2-inch pipe has an area of 0.0233 square foot, therefore the velocity must be about 17.6 feet per minute, or 0.293 foot per second.

The velocities in the pipes of hot water heating systems are very low, as they must be, for the water to circulate with the small head, due to the difference in temperature between the water in the flow and return pipes.

RADIATING SURFACE SUPPLIED BY PIPES OF DIFFERENT SIZES.

If the volume of water passing through pipes of different sizes is known, the radiating surface they will supply may be readily computed. With the same drop in temperature in radiators, the force causing circulation will be alike in all. With pipes of equal length the resistance will vary as the square of the velocity, and

inversely as for the diameter expressed, as $\frac{v^2}{d}$.

Now, if we assume, for example, that a 2-inch pipe will supply 200 square feet of direct radiation—which in practice it will readily do—and compute the value of $\frac{v^2}{d}$, then make $\frac{v^2}{d}$ the same for pipes of other sizes, a table may be prepared showing the radiating surface that may be supplied by pipes of different diameters when working under the same conditions with respect to the head causing the flow and the resistance to the circulation. This has been done, and the results are stated in the following table:

TABLE XIX.

THE CAPACITY OF MAINS 100 FEET LONG EXPRESSED IN THE NUMBER OF SQUARE FEET OF DIRECT HOT WATER RADIATING SURFACE THEY WILL SUPPLY WITH THE OPEN TANK SYSTEM, WHEN THE RADIATORS ARE PLACED IN ROOMS AT 70 DEGREES F.

Nominal diameter of pipes. Inches.	Capacity in square feet of direct radiating surface.	Actual inside diameter in inches.	Actual inside diameter in feet.	Area in square inches.	Area in square feet.	Capacity in gallons per foot length.
1¼	75	1.38	0.125	1.49	0.0104	0.0777
1½	107	1.61	0.134	2.04	0.0141	0.1058
2	200	2.07	0.172	3.35	0.0233	0.1743
2½	314	2.47	0.206	4.78	0.0332	0.2483
3	540	3.07	0.256	7.39	0.0513	0.3835
3½	780	3.55	0.296	9.89	0.0687	0.5136
4	1,060	4.03	0.333	12.73	0.0884	0.6613
4½	1,410	4.50	0.375	15.94	0.1108	0.829
5	1,860	5.04	0.417	19.99	0.1388	1.038
6	2,960	6.06	0.500	28.89	0.2006	1.500
7	4,280	7.02	0.583	38.74	0.2690	2.012
8	5,850	7.98	0.666	50.04	0.3474	2.599

NOTE.—The above ratings in the second column are based on buildings having not more than three floors above the basement. With higher buildings the capacities would be increased.

It is of some interest to compare with Table XIX the pipe capacities that have been presented in various publications and trade catalogues. Table XX gives such a comparison, and shows a wide variation in the computed capacities stated by various engineers:

TABLE XX.

THE CAPACITY OF HOT WATER HEATING MAINS EXPRESSED IN THE NUMBER OF SQUARE FEET OF DIRECT RADIATING SURFACES SUPPLIED.

Diameter of pipe. Inches.	A.	B.	C.	D.	E.	F.	G.
1	39	44	....	30	50	....	89
1¼	64	69	78	60	90	....	140
1½	95	100	111	100	130	200	202
2	156	176	184	200	250	325	359
2½	256	275	260	350	400	450	561
3	381	400	405	550	540	700	807
3½	531	540	576	850	740	900	1,099
4	706	710	784	1,200	890	1,200	1,436
4½	906	890	990	....	1,100	1,500	1,817
5	1,131	1,100	1,240	....	1,600	2,000	2,244
6	1,525	1,600	1,920	....	....	3,000	3,228
7	2,150	....	2,760	....	....	4,200	4,396
8	2,750	....	3,570	....	....	5,600	5,744
9	3,625	....	....	....	....	....	7,268
10	4,525	....	6,050	....	....	....	8,976

Authorities.

A—J. L. Bixley; B—J. H. Kinealy; C—J. L. Mott Iron Works; D—C. L. Hubbard; E—R. C. Carpenter; F—Model Heating Company; G—Nason Mfg. Company.

PIPE SIZES FOR INDIRECT HEATING.

Since indirect radiators are placed at a much lower level, with reference to the heater, than are direct radiators, the head corresponding to the difference in temperature between the supply and the return pipes is much less than is the case with the latter, and scarcely exceeds 1-20 foot. Since cold air comes in contact with the radiators, the loss of heat per square foot is much greater than from direct radiators. These two causes make it necessary to provide much larger pipes to supply a given amount of surface than in the case of direct radiators.

For supplying indirect radiators, C. L. Hubbard recommends using 1¼-inch pipes for 30 square feet, 1½-inch for 31 to 50, 2-inch for 51 to 100, 2½-inch for 101 to 200, 3-inch for 201 to 300, 3½-inch for 301 to 400, and 4-inch for 401 to 600. Baldwin recommends allowing a 2-inch pipe to 100 square feet of indirect radiation. This rule gives much larger pipes than customary. Certain hot water fitters use 1¼-inch pipes to 60 square feet, 1½-inch for 61 to 120, and 2-inch for 121 to 240 square feet. With pipes carrying so much radiating surface as the latter, the drop in temperature of the water in passing through the radiators must

be greater than when larger pipes are used. The objection to small pipes, with consequent increased drop in temperature to overcome resistance, is that the mean temperature of the radiator is lowered, and the heat given off per square foot is diminished. What is saved in piping must be made up in radiation.

The writer considers it unwise to supply more than 200 square feet of indirect radiation with a 2-inch pipe, and prefers rating a 2-inch pipe to supply 150 square feet of indirect surface. Taking the latter as a basis, pipes of other sizes would supply the surface stated in Table XXI when working against the same resistance, which varies as the square of the velocity and inversely as the diameter.

TABLE XXI.

THE CAPACITIES OF PIPES EXPRESSED IN THE NUMBER OF SQUARE FEET OF INDIRECT HOT WATER RADIATING SURFACE THEY WILL SUPPLY WITH THE OPEN TANK SYSTEM.

Diameter of pipes. Inches.	Indirect radiating surface. Square feet.	Diameter of pipes. Inches.	Indirect radiating surface. Square feet.
1¼	56	4	790
1½	80	4½	1,060
2	150	5	1,400
2½	235	6	2,220
3	405	7	3,200
3½	585	8	4,400

SIZES OF RISERS.

The capacities of risers recommended by different writers are as follows:

TABLE XXII.

COMPARISON OF RATINGS FOR HOT WATER RISERS.
(Hight of floors approximately 10 feet each.)

Sizes of pipes in inches.	First-floor risers. Square feet direct radiation.				Second-floor risers. Square feet direct radiation.			
¾	...	27	...	50	...	35	...	52
1	39	48	30	89	45	62	55	92
1¼	64	75	60	140	73	97	90	144
1½	95	108	100	202	110	140	140	209
2	156	191	200	359	179	250	275	370
2½	256	300	350	561	294	390	475	577
3	381	430	550	807	438	...	...	835
3½	531	590	850	1,099	610	...	...	1,132
4	706	770	...	1,436	812	...	...	1,478
4½	906	970	...	1,817	1,042	...	...	1,871
5	1,131	1,200	...	2,244	1,301	...	...	2,309
6	1,525	1,700	...	3,228	1,753	...	...	3,341

TABLE XXII—Continued.

	Third-floor risers. Square feet direct radiation.				Fourth-floor risers. Square feet direct radiation.			
¾	...	35	...	53	...	...	...	55
1	48	62	65	95	52	...	75	98
1¼	79	97	110	149	85	...	125	153
1½	118	140	165	214	126	...	185	222
2	194	250	375	380	206	...	425	393
2½	318	390	...	595	338	...	...	613
3	473	...	...	856	503	...	...	888
3½	659	...	...	1,166	701	...	...	1,202
4	876	...	...	1,520	932	...	...	1,571
4½	1,124	...	...	1,927	1,196	...	...	1,988
5	1,402	...	...	2,376	1,493	...	...	2,454
6	1,891	...	...	3,424	2,013	...	...	3,552

The figures stated in the first, second, third and fourth columns, giving capacities, are by Bixley, Kinealy, Hubbard and Nason, respectively. It will be noted that here, as in the case of mains, the capacities given by the Nason Company are much in excess of others. The figures given by Prof. Kinealy are based on water at high temperature and may be increased 25 per cent. for water at 160 degrees in the radiator.

The following table has been compiled by the writer, using as a basis a 1½-inch pipe rated to supply 100 square feet of direct radiation on the first floor, 140 square feet on the second, 175 square feet on the third and 200 square feet on the fourth. The capacities of other pipes are based on a flow that represents the same resistance to be overcome as in the 1½-inch pipes, as above rated; that is, the capacities of pipes larger than 1½-inch are based on a higher velocity and smaller pipes on a correspondingly lower velocity, since the resistance varies directly as the square of the velocity and inversely as the diameter.

TABLE XXIII.

THE CAPACITIES OF RISERS EXPRESSED IN THE NUMBER OF SQUARE FEET OF DIRECT HOT WATER RADIATING SURFACE THEY WILL SUPPLY ON DIFFERENT FLOORS.—FLOOR HIGHTS APPROXIMATELY 10 FEET.—OPEN TANK SYSTEM. —RADIATORS IN ROOMS AT 70 DEGREES F.

Diameter of riser. in inches.	Square feet of direct radiating surface supplied. First floor.	Second floor.	Third floor.	Fourth floor.
1	33	46	57	64
1¼	71	104	124	142
1½	100	140	175	200
2	187	262	325	375
2½	292	410	492	580
3	500	755	875	1,000

RADIATOR CONNECTIONS.

Direct hot water radiators are commonly tapped 1 inch up to 40 square feet, 1¼ inches for 41 to 72 square feet, and 1½ inches for ordinary sizes larger than 72 square feet.

ELBOWS AND BENDS.

The resistance interposed by elbows to the passage of water is a subject on which there appears to be little available data of value. Fortunately, it is unnecessary, in ordinary heating work, to compute the loss of heat due to this resistance. The writer, in a series of articles on the flow of steam, gives a table showing the lengths of straight pipe that would present the same resistance as a standard elbow. The values there given will be found convenient for use in case it is desired to allow for the resistance of elbows in an extensive system of hot water heating.

In the smaller sizes of fittings, say from 1½ to 4 inches, the radius of the center line of the elbow is roughly 1¼ x the diameter of pipe for standard elbows; 1¾ x the diameter of pipe for the long turn patterns and 2¼ x the diameter of pipe for extra long turn elbows. The relative resistance, or loss of head, computed from Weisbach's formula is, for these three patterns, as follows: Standard, 100; long turn, 83; extra long turn, 77. While these figures may be considered merely approximate, they serve to show in a general way the great advantage of long turn elbows over those of standard patterns for hot water work.

Ordinary wrought iron or steel pipe bends have a radius of axis equal to, at least, 5 x the diameter of pipe. With such bends, Trautwine states, the flow will not be materially diminished. In first-class hot water heating plants long turn elbows are used, and the ends of the pipes are reamed inside to reduce, as far as possible, the resistance to the flow of water and to permit the least difference possible between the temperature in the supply and return pipes.

EXPANSION TANKS.

Hot water expands about 4 per cent. of its volume at 40 degrees when heated to 200 degrees. Taking these as the extremes of temperature between the water when the system is first filled and when operating in coldest weather, and assuming that the expansion tank should have a capacity equal to twice this increase in volume, the tank should be made 8 per cent., or about one-twelfth, of the total volume of radiator and piping. Suppose the piping is equivalent to one-third the direct radiating sur-

face and the volume of water in the system to amount to 1½ pints per square foot of radiating surface, including mains, then, for example, a 10-gallon expansion tank would be adapted to a system holding 120 gallons, which, on the basis of 1½ pints per square foot of radiation, mains included, would be 640 square feet. And, since mains are reckoned at one-third the actual surface in radiation, the latter would amount to three-fourths of 640 square feet equal 480 square feet, or, say, in round numbers, 500 square feet. On the same basis the capacity of other tanks would be in proportion, as follows:

TABLE XXIV.

CAPACITY OF EXPANSION TANKS.

Capacity of tank in gallons.	Capacity in square feet of actual surface in hot water radiator to which tank is adapted.
5	250
10	500
15	750
20	1,000
30	1,500
40	2,000
50	2,500
60	3,000

It will be noted that the capacities in the above table are equivalent to 1 gallon in expansion tank to each 50 square feet of surface in radiators; a convenient rule. While tanks may be made smaller, the saving would be slight, and they would require more frequent attention, unless fitted with an automatic water line regulator.

It is beyond the scope of this work to discuss methods of piping; yet the writer feels constrained to warn fitters against the danger in placing a valve in the expansion pipe, which is sometimes done, and also to see to it that the expansion pipe and tank are located where there will be no danger from freezing.

CHAPTER VIII.

THE FLOW OF STEAM IN PIPES AND THE CAPACITIES OF PIPES FOR STEAM HEATING SYSTEMS AND FOR STEAM BOILERS.

The following chapter is intended not as an exhaustive discussion of the various methods of proportioning piping systems, nor of the formulas on which the flow of steam is based, but to provide, by tables, a ready means of solving problems relating to pipe sizes. The formulas on which the tables are based make due allowance for the diminished resistance due to an increase in the size of pipes.

The cruder, yet common, method of allowing, for large and small pipes alike, a certain number of thousandths of a square inch in cross sectional area to each square foot of radiating surface makes the larger pipes much greater in area in proportion to the surface supplied than the smaller ones.

A COMPARISON OF FORMULAS.

D'Arcy's modified formula, stated in Kent's "Mechanical Engineer's Pocketbook," gives the weight of steam that will flow per minute through pipes of various sizes as

$$W = c\sqrt{\frac{w\,(p_1 - p_2)\,d^5}{L}} \quad \text{..................} (a)$$

where $w =$ the density or weight per cubic foot of steam at pressure p_1; $(p_1 - p_2) =$ drop in pressure, or the difference between initial and terminal pressure; $d =$ diameter of pipe in inches; L $=$ length of pipe in feet; $c =$ coefficient, as follows:

Diameter of pipe in inches. 1	2	3	4	5	6	7	8	9
Value of c..................45.3	52.7	56.1	57.8	58.4	59.5	60.1	60.7	61.2
Diameter of pipe in inches. 10	11	12	14	16	18	20	22	24
Value of c..................61.8	62	62.1	62.3	62.6	62.7	62.9	63.2	63.2

Babcock's formula, given in "Steam," is

$$W = 87\sqrt{\frac{w\,(p_1 - p_2)\,d^5}{L\left(1 + \frac{3.6}{d}\right)}} \quad \text{..................} (b)$$

This may be reduced to a form similar to D'Arcy's formula, but with different coefficients.

Table XXV has been computed from these formulas in order to compare the results for pipes of different sizes. This table is based on the actual inside diameter of standard wrought iron pipes of nominal sizes stated up to 12 inches, inclusive. Sizes of 14 inches and larger are nominal outside diameters of O. D. pipe, the inside diameter of each being ¾-inch less than the outside.

TABLE XXV.

SHOWING THE WEIGHT OF STEAM IN POUNDS THAT WILL FLOW PER MINUTE THROUGH STRAIGHT PIPES 100 FEET IN LENGTH; NO ALLOWANCE BEING MADE FOR RESISTANCE AT THE ENTRANCE TO THE PIPE.

Initial pressure, 5 pounds by gauge, less a drop in pressure in a length of 100 feet, 1 pound.

Formula.	Nominal diameter of pipe in inches.											
	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7
	Weight of steam, in pounds, flowing through pipe per minute.											
D'Arcy's	1.14	2.38	3.7	6.7	11.6	20.8	30.3	41.4	56	73	118	174
Babcock's	1.05	2.31	3.6	7.3	11.9	21.9	32.7	46.5	63	86	141	208

Formula.	Nominal diameter of pipe in inches.									
	8	9	10	12	14	16	18	20	22	24
	Weight of steam, in pounds, flowing through pipe per minute.									
D'Arcy's	246	327	438	694	910	1,266	1,735	2,285	2,945	3,660
Babcock's	293	394	533	853	1,140	1,590	2,210	2,910	3,760	4,730

APPLICATION OF FACTORS TO TABLE XXV.

Both formulas show the weight of steam delivered to be proportional: (1) To the square root of the density or the square root of the weight per cubic foot; (2) to the square root of the drop in pressure; (3) to the square root of the fifth power of the diameter of the pipe, and (4) to be inversely proportional to the square root of the length of the pipe.

For any other pressure than 5 pounds, on which Table XXV is based, multiply the figures there stated by the square root of the density at the given pressure, divided by the square root of the density at 5 pounds pressure. This gives the following factors for different pressures:

TABLE XXVI.

FACTORS TO BE APPLIED TO TABLE XXV FOR OTHER GAUGE PRESSURES THAN 5 POUNDS.

Gauge pressure in pounds	1	2	5	10	15	20	30	40
Multiplier	0.90	0.93	1.00	1.11	1.21	1.31	1.47	1.61
Gauge pressure in pounds	50	60	70	80	90	100	110	120
Multiplier	1.74	1.86	1.97	2.07	2.18	2.26	2.37	2.46

This table shows, for example, that with 50 pounds pressure 1.74 times as much steam by weight will flow through a given pipe as with 5 pounds pressure; the drop in pressure being the same in each case.

For a drop in pressure other than 1 pound, on which Table XXV is based, multiply the figures in that table by the square root of the given drop corresponding to these factors.

TABLE XXVII.

FACTORS APPLYING TO TABLE XXV FOR OTHER DROPS IN PRESSURE THAN 1 POUND.

Drop in pressure in pounds ($p_1 - p_2$)	⅛	¼	½	¾	1	2	3
Multiplier	0.354	0.500	0.709	0.865	1.00	1.41	1.73

TABLE XXVIII.

FACTORS FOR OTHER LENGTHS THAN 100 FEET. TOTAL DROP IN PRESSURE ASSUMED TO BE 1 POUND, WHATEVER THE LENGTH OF THE PIPE THE CAPACITY OF WHICH IS BEING COMPUTED. FACTORS OR MULTIPLIERS TO BE USED IN CONNECTION WITH TABLE XXV.

Length of pipe in feet	50	100	150	200	250	300	350	400	450
Multiplier	1.41	1.00	0.816	0.710	0.632	0.578	0.535	0.500	0.471
Length of pipe in feet	500	550	600	650	700	750	800	850	
Multiplier	0.447	0.427	0.407	0.392	0.379	0.365	0.353	0.343	
Length of pipe in feet	900	950	1,000	1,200	1,400	1,600	1,800	2,000	
Multiplier	0.333	0.325	0.316	0.288	0.268	0.250	0.236	0.224	

To illustrate the use of Tables XXV, XXVI, XXVII and XXVIII, suppose it is desired to compute the flow of steam at 50 pounds gauge pressure through a 3-inch pipe 400 feet long, the drop in pressure to be 2 pounds. Table XXV gives, with D'Arcy's formula, 20.8 pounds as the weight of steam passing in one minute through a pipe 100 feet long, with 1 pound drop in pressure. Applying the factors in Tables XXVI, XXVIII and XXVII, respectively, corresponding to the above conditions, we have $20.8 \times 1.74 \times 0.5 \times 1.41 = 25.43$ pounds as the weight of steam flowing through the pipe per minute.

RESISTANCES TO THE FLOW OF STEAM.

In computing the flow of steam from Table XXV, the resistance at the entrance to the pipe at elbows and globe valves should be allowed for by adding to the actual length of the pipe a length that would produce the same resistance to the flow as that at these

points. The resistance at the entrance is commonly expressed in connection with Babcock's formula by the equation

$$R = \frac{114 \textit{ diameters}}{1 + (3.6 \div d)} \ldots \text{“}c.\text{”}$$

where R equals a length of straight pipe expressed in diameters that would interpose the same resistance as that at the entrance and d equals diameter of pipe in inches.

Very little has been published giving the results of tests bearing on this subject. Treatises on hydraulics, in discussing the flow of water in pipes, which follows the same general laws as the flow of steam, give tables and data showing the length of pipe equivalent in resistance to that at entrance to be approximately one-third of that given by formula "*c*." The use of formula "*c*" in computing pipe sizes for steam heating systems gives sizes much in excess of those found necessary in practice. The author, therefore, favors the use of

$$R = \tfrac{1}{3} \times \frac{114 \textit{ diameters}}{1 + (3.6 \div d)} \ldots\ldots \text{“}d.\text{”}$$

The values corresponding to the latter formula, reduced to feet, are as follows:

TABLE XXIX.

THE RESISTANCE AT THE ENTRANCE OF PIPES EXPRESSED IN THE NUMBER OF FEET OF STRAIGHT PIPE THAT WOULD PRODUCE THE SAME RESISTANCE AS THAT AT THE ENTRANCE.

*Nominal diameter of pipes in inches.	Resistance based on Formula "*d.*"—Feet.	*Nominal diameter of pipes in inches.	Resistance based on Formula "*d.*"—Feet.
1	0.8	7	14.7
1¼	1.2	8	17.5
1½	1.6	9	20.4
2	2.4	10	23.4
2½	3.1	12	29.4
3	4.5	14	35.3
3½	5.1	16	41.3
4	6.7	18	47.3
4½	7.9	20	53.6
5	9.3	22	60.0
6	12.1	24	66.0

* Nominal diameter of 14-inch pipes and larger is the outside diameter.

The resistance at a globe valve of given size is commonly allowed for by adding to the actual length of pipe a length three times that stated in Table XXIX., and for a standard elbow a length twice that stated in the table. These values are, however,

to be considered as only approximately true, although they are near enough for practical use. The longer the pipe the less will be the error in the computed flow due to any uncertainty or error in the allowance made for the three resistances at entrance, elbows and globe valves.

The use of long turn elbows and straightway gate valves practically eliminates two of these resistance losses, and the other is considerably reduced by reaming the pipes at the ends, as is common in hot water work.

The following example will illustrate the use of Table XXV, the multipliers in Tables XXVI, XXVII, and XXVIII, and allowance in Table XXIX.

How many pounds of steam will flow per minute through a 4½-inch pipe 800 feet long, with four elbows and one globe valve? Initial gauge pressure, 10 pounds. Drop in pressure, 2 pounds.

	Feet.
Actual length of pipe	800
Allowance for loss at entrance (Table XX approximately)	8
Allowance for two elbows	32
Allowance for one globe valve	24
Total equivalent length of straight pipe, making due allowances as above.	864

A length of 850 feet is the one most nearly corresponding to this length in Table XXVIII. The factor for this length is 0.343; for 10 pounds pressure the factor in Table XXVI is 1.11; for 2 pounds drop in pressure the factor in Table XXVII is 1.41. The flow of steam in pounds by D'Arcy's formula, stated in Table XXV, to which these factors apply, is for a 4½-inch pipe 56 pounds. For a length of 800 feet, with conditions as stated, the computed flow would be $56 \times 0.343 \times 1.11 \times 1.41 = 30.1$ pounds per minute.

EFFECT OF CONDENSATION.

No account has been taken in the foregoing of the effect of condensation on the flow of steam. It is assumed that pipes will be covered, which will reduce this effect to about one-third of what it would be if they were bare. The condensation, while it cuts down the volume of steam, at the same time causes a greater drop in pressure. This, in turn, increases the velocity, tending to

offset the loss by condensation. A further discussion of this subject may be found in Kent's "Mechanical Engineers' Pocket Book."

STEAM FLOW WITH MORE THAN 40 PER CENT. DROP IN PRESSURE.

It is to be borne in mind, in making computations of the flow of steam, that steam of 25 pounds gauge pressure or more, discharged from a pipe against atmospheric pressure or against a pressure less than three-fifths the initial pressure, has a nearly constant velocity of 900 feet per second, in round numbers, the weight discharged increasing with the pressure and being proportional to the density or weight per cubic foot. The approximate weight of steam that will flow per hour through a pipe under the conditions just stated is equal to 50 × (absolute pressure of steam) × (area of pipe in square inches).

This constant velocity applies only to very short pipes.

RELATIVE CAPACITIES OF PIPES.

The relative capacities of pipes under the same conditions are shown in Table XXV, D'Arcy's values being the safer to use. This table will be found convenient in determining the size of pipe necessary to supply a number of smaller ones.

Example: What size pipe is required to supply one 2½, two 3 and one 4 inch pipes?

The capacities in Table XXV are, in the order stated, 1 x 11.6, 2 x 20.8, 1 x 41.4; total, 94.6. A 6-inch pipe with a capacity of 118 should be used, since a 5-inch pipe has a capacity of only 73.

It will be noted that the capacity of pipes increases much more rapidly than their area—*e. g.*, the relative capacities of 8 and 4 inch pipes in Table XXV are 246 and 41.4, or in the ratio of 6 to 1, whereas their areas are in the ratio of about 4 to 1.

Table XXX, which has been computed from Table XXV, gives the proper size of mains to supply branches of the sizes stated in the upper and side lines.

TABLE XXX.

EQUATION OF PIPES.

	Size mains to supply branches of sizes stated in upper and side lines.																
Size branch	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	Size branch
	1¼	1½	2	2½—	3—	3+	3½+	4+	4½+	5+	6+	7+	8+	9+	10+	12+	1
	..	2—	2—	2½—	3—	3½—	3½+	4+	4½+	5+	6+	7+	8+	9+	10+	12+	1¼
	..	..	2½—	2½—	3—	3½—	4—	4½—	4½+	5+	6+	7+	8+	9+	10+	12+	1½
	..	..	..	3—	3½—	3½—	4—	4½—	5—	6—	6+	7+	8+	9+	10+	12+	2
	..	..	..	..	3½—	4—	4—	4½—	5—	6—	7—	8—	8+	9+	10+	12+	2½
	..	..	..	..	..	4	4½—	5—	5+	6—	7—	8—	9—	10—	10+	12+	3
	..	..	..	..	..	..	5	5—	6—	6—	7—	8—	9—	10—	12—	12+	3½
	..	..	..	..	..	..	..	6—	6—	6—	7—	8—	9—	10—	12—	12+	4
	..	..	..	..	..	..	..	..	..	7—	8—	8	9—	10—	12—	14—	5
	..	..	..	..	..	..	..	..	..	..	8—	9—	9+	10+	12—	14—	6
	..	..	..	..	..	..	..	..	..	..	..	9—	10—	12+	12—	14—	7
	..	..	..	..	..	..	..	..	..	..	..	..	12+	12—	12—	14—	8
	..	..	..	..	..	..	..	..	..	..	..	..	..	12—	12+	14—	9
	..	..	..	..	..	..	..	..	..	..	..	..	..	..	14—	15—	10
	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	16	12

NOTE.—Minus signs (—) indicate that mains of the sizes stated are of ample size to supply the branches stated opposite them. Plus signs (+) indicate that mains of the sizes stated are a trifle smaller than given by the formula, but not sufficiently so to warrant increasing the main to the next size, except when the requirements are unusually exacting as to the permissible drop in pressure.

TABLE XXXI.

THE AMOUNT OF ORDINARY CAST IRON RADIATING SURFACE THAT MAY BE SUPPLIED BY PIPES OF DIFFERENT SIZES, 100 FEET LONG, WITH 5 POUNDS INITIAL GAUGE PRESSURE AND THE DROP IN PRESSURE STATED IN THE COLUMN AT THE LEFT. FOR OTHER PRESSURES AND FOR LENGTHS IN EXCESS OF 100 FEET, USE FACTORS IN TABLES XXVI AND XXVIII. RESISTANCE AT ENTRANCE, ELBOWS AND GLOBE VALVES MAY BE ALLOWED FOR AS STATED IN TABLE XXIX, BUT THIS IS GENERALLY UNNECESSARY FOR ORDINARY WORK, A SLIGHT EXCESS IN THE DROP IN PRESSURE OVER THAT ASSUMED COMPENSATING FOR THE RETARDING EFFECT OF THE ENTRANCE, ELBOWS AND VALVES.

Diameter of pipes, Inches........		1	1¼	1½	2	2½	3	3½	4	4½
Line.	Drop in pressure. Pounds.	Square feet of radiating surface.								
A............	1	261	545	847	1,535	2,660	4,770	6,950	9,500	12,820
B............	¾	226	472	732	1,325	2,300	4,120	6,000	8,210	11,100
C............	½	185	386	600	1,087	1,881	3,370	4,930	6,730	9,100
D............	¼	130	273	423	767	1,330	2,385	3,475	4,750	6,210
E............	⅛	92	193	299	543	940	1,680	2,460	3,360	4,530
F............	1/16	65	136	212	384	665	1,192	1,740	2,380	3,210
G............	1/32	46	96	150	272	470	845	1,230	1,680	2,270

Diameter of pipes, inches........		5	6	7	8	9	10	12
Line.	Drop in pressure. Pounds.	Square feet of radiating surface.						
A............	1	16,710	27,000	39,800	56,300	75,000	100,000	159,000
B............	¾	14,450	23,350	34,400	48,750	64,800	86,500	137,500
C............	½	11,810	19,100	28,200	39,800	53,000	70,800	112,500
D............	¼	8,355	13,500	19,900	28,150	37,400	50,000	74,500
E............	⅛	5,900	9,530	14,060	19,900	26,400	35,400	56,200
F............	1/16	4,180	6,750	9,950	14,100	18,700	25,000	39,800
G............	1/32	2,960	4,780	7,050	10,000	13,300	17,700	28,200

Diameter of pipe, inches........		14	16	18	20	22	24
Line.	Drop in pressure. Pounds.	Square feet of radiating surface.					
A............	1	214,000	290,000	398,000	524,000	675,000	845,000
B............	¾	186,000	250,000	343,000	453,000	583,000	730,000
C............	½	151,000	206,000	282,000	371,000	478,000	598,000
D............	¼	107,000	145,000	198,000	262,000	338,000	425,000
E............	⅛	75,100	102,500	140,000	185,000	238,000	298,000
F............	1/16	53,300	72,600	99,000	131,000	169,000	212,000
G............	1/32	37,900	51,300	70,500	93,000	......	......

NOTE.—Sizes 14 inches and larger are outside diameters. For sizes of returns see note under Table XXXIV.

SIZES OF STEAM HEATING MAINS.

For steam heating work it is generally more convenient to deal with heat units and the amount of direct rediating surface that pipes of different sizes will supply, than with the weight of steam they will carry. A conservative basis is to allow 250 heat units per hour per square foot of ordinary cast iron direct radiating surface, with steam at low pressure, say 3-5 pounds.

A pound of steam at 5 pounds pressure has a latent heat of 954.6 units—that is, it will give up 954.6 heat units when condensed to water at the steam temperature in the radiator.

Table XXXI has been deduced from the flow of steam computed from D'Arcy's formula, as stated in Table XXV, on the basis of 250 heat units per square foot of radiation and 954.6 heat units given off per pound of steam, which is equivalent to 0.262 pounds of steam condensed per hour per square foot of direct radiating surface.

Table XXXI shows a marked difference in the amount of radiating surface that may be applied with different assumed drops in pressure.

Mains may be proportioned as follows:

For systems trapped to an open receiver with the heating surface located well above same, a drop of ¼ to ½ pound may be allowed.

For gravity return systems, where the radiators are located some distance, say 5 feet or more, above the water line in the boiler, 1-16 to ⅛ pound drop may be assumed in proportioning the piping. Where the radiators are but little above the water line, as in indirect systems, use 1-32 pound drop.

When exhaust steam is used and it is desired that the minimum back pressure be carried on the system, an assumed drop of 1-32 to 1-16 pound may be used, preferably 1-32 pound drop.

The size of vertical pipes or overhead feed risers of single pipe systems may be based on line G, Table XXXI. This will give sizes corresponding to those based on 2 pounds pressure with a little greater drop than 1-32 pound, and will be found ample for exhaust steam heating.

In high buildings, with the single pipe overhead feed system, the risers must be very liberally proportioned on the lower stories, since they must carry not only steam to the radiators below, but the condensation from the radiators above.

It should be noted that pipe sizes based on the recommendations just made will be large enough to supply steam at, say, 2 or 3 pounds pressure to the radiating surface stated, but with a slightly greater drop in pressure.

Pipe sizes given in Table XXXI to supply a given radiating surface with steam at 5 pounds pressure will be very nearly correct

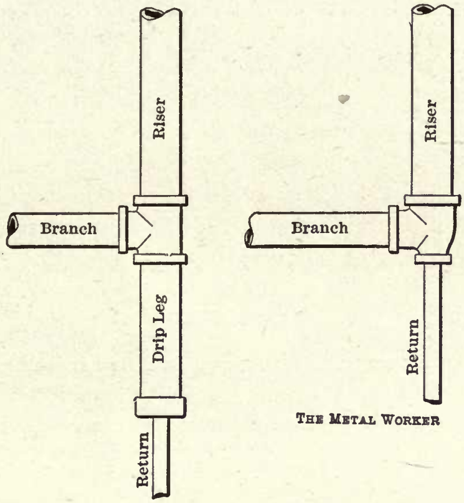

Fig. 34. Fig. 35.

for higher pressures within the ordinary limits of steam heating, say up to 15 pounds. This is true, since the total heat supplied by the steam at higher pressures, taking into account its increased weight and decreased latent heat just about keeps pace with the increased radiation of heat.

Direct steam radiators are commonly tapped as shown in the following table:

TABLE XXXII.

DIRECT RADIATOR TAPPING.

ONE-PIPE SYSTEMS.

Up to 24 square feet inclusive	1 inch
24– 60 "	1¼ "
60–100 "	1½ "
Over 100 "	2 "

TWO-PIPE WORK

Up to 48 square feet inclusive	1 x ¾ inch
48–96 "	1¼ x 1 "
Over 96 "	1½ x 1¼ "

SIZES OF RISERS—ONE-PIPE SYSTEM.

The risers of one-pipe up feed steam heating systems must be made large enough to supply the radiators and also permit the condensation to return by the same route. It is, therefore, well to limit the velocity to, say, 15 feet per second. On this basis, with steam of 2 pounds pressure, pipes will supply ordinary cast iron direct radiators as follows:

TABLE XXXIII.

CAPACITY OF UP FEED RISERS, ONE-PIPE SYSTEM.

Size of riser, one-pipe system.—Velocity steam, 15 feet per second.	Approximate radiating surface supplied.—Steam, 2 pounds pressure.	Size of riser, one-pipe system.—Velocity steam, 15 feet per second.	Approximate radiating surface supplied.—Steam, 2 pounds pressure.
Inches.	Feet.	Inches.	Feet.
1	*50	2½	300
1¼	*90	3	460
1½	130	3½	620
2	210	4	800

* It is advisable to make the upper end of riser the full size of standard radiator connections—viz., 1 inch up to 24 feet and 1¼ inches for 25 to 60 feet.

Down feed risers may be safely rated to carry at least 25 per cent. more surface than stated in the table. Care should be taken to proportion the risers liberally near the lower end to provide for the removal of condensation.

Branch connections with radiators should be the same size as regular tapping, except when radiators are located more than 4 or

5 feet from risers. In this event the connections should be increased one size to lessen the velocity and permit the condensation to easily flow back against the current of steam.

It is better to drip the riser as indicated in Fig. 34 than as shown in Fig. 35. With the latter the condensation is apt to be swept up along the heel of the elbow, causing a click, or water hammer. The arrangement shown in Fig. 34 forms a separator and the condensation trickles away quietly through the relief or return pipe.

SIZES OF RISERS—TWO-PIPE SYSTEM.

With the two-pipe up feed system risers may be considerably smaller for a given radiating surface than in the one-pipe system, since the condensation from the radiators is conducted away through separate returns.

Safe allowances are given in the following table:

TABLE XXXIV.

CAPACITIES OF UP FEED RISERS, TWO-PIPE SYSTEM.

Size of riser. for two-pipe, up feed steam heating. Inches.	Approximate radiating surface supplied.—Steam at 2 pounds pressure. Feet.	Size of riser for two-pipe up feed steam heating. Inches.	Approximate radiating surface supplied.—Steam at 2 pounds pressure. Feet.
1	*70	2½	570
1¼	*130	3	1,020
1½	*190	3½	1,490
2	330	4	2,000

* It is advisable, at the upper ends of long risers, to make the riser the full size of standard radiator connections—viz., 1 inch up to 48 feet; 1¼ inches for 49 to 96 feet, and 1½ inches for 97 and up to 190 feet.

For down feed risers it is safe to allow 25 per cent. more surface than stated in the above table.

In high buildings, say over five or six stories, allow 10 per cent. less surface than that stated to allow for the increased length of and condensation in risers. Returns are commonly made one size smaller than the supply up to 2½ inches; above that the returns may be two sizes smaller, and for larger pipes, where the return has ample fall, it may be made one-half the diameter of the supply pipe, or even smaller.

THE ONE-PIPE CIRCUIT SYSTEM.

For the one-pipe circuit system of piping, that is, where a loop is run in the basement from which branches lead to the supply risers, there to drain to the circuit main, this in turn being dripped at the end, the author recommends the following capacities.

These capacities are based on all condensation, both from radiators and risers, being carried through the circuit main.

TABLE XXXV.

CIRCUIT SYSTEM—PIPE SIZES AND APPROXIMATE SAFE CAPACITIES.

2	inch diameter	300 square feet direct radiation
2½	"	500 " "
3	"	900 " "
3½	"	1,200 " "
4	"	1,700 " "
4½	"	2,300 " "
5	"	3,000 " "
6	"	5,000 " "
7	"	7,500 " "
8	"	10,000 " "

Where conditions are favorable, circuit mains comparatively short, and may have a good pitch, these capacities may be materially increased.

PIPE SIZES FOR THE TWO-PIPE VACUUM SYSTEM OF STEAM HEATING.

Supply connections with radiators and coils in the two-pipe vacuum system of steam heating are commonly ¾ inch up to 50 square feet, 1 inch for 51 to 100 square feet, 1¼ inches for 101 to 190 square feet, 1½ inches for 191 to 310 square feet, 2 inches for 311 to 670 square feet, 2½ inches for 671 to 1250 square feet and 3 inches for 1251 to 2040 square feet. It will be noted that these are considerably smaller than pipe connections with ordinary low pressure heating systems.

Sizes of up feed risers in buildings of six or eight stories may be based on Table XXXI, line D. In proportioning risers in high office buildings with the down feed vacuum system Table XXXI, line E, may be used. The lower portion of such risers should be proportioned to supply 10 to 15 per cent. less surface than that stated in the table, since they must not only supply steam to the radiators, but must carry away the condensation from the attic mains and

from the risers above. Return risers are very much smaller than with the ordinary two-pipe system.

The following table gives safe allowances for horizontal returns. These allowances may be safely increased 20 to 30 per cent. for short lines.

TABLE XXXVI.

RETURN PIPE CAPACITIES FOR TWO-PIPE VACUUM SYSTEMS.

Size of return pipe.	Square feet of direct radiating surface to which return pipe is adapted.
¾ inch	200
1 inch	200- 800
1¼ inches	800- 1,500
1½ inches	1,500- 3,000
2 inches	3,000- 6,000
2½ inches	6,000-12,000
3 inches	12,000-20,000
3½ inches	20,000-30,000
4 inches	30,000-40,000

Return risers may be rated as follows:

¾ inch up to	1,000 feet.
1 inch up to	1,600 feet.
1¼ inches up to	2,500 feet.

The size of main steam supply pipes and branches should be based on the distance of the most remote radiator from the source of supply, the distance in overhead feed systems to be measured from the center of distribution in the attic. It is not necessary to add the length of the main exhaust pipe from the basement, since it is merely a reservoir of steam, and distribution really begins at the tee connected with the same in the attic.

For lengths of 100 feet or less pipe sizes based on line C of Table XXXI agree fairly closely with common practice in this class of work. For other lengths, multiply the figures in line C by the factors in Table XXVIII in order to ascertain the capacity of pipes, expressed in the square feet of radiating surface, they will supply.

Smaller pipes, both for supply and return, could be made to do the work by carrying a few ounces more pressure on the mains or by causing the pumps to maintain a stronger pull on the returns.

COMPARISON OF DIFFERENT METHODS OF DETERMINING THE SIZE OF STEAM MAINS TO SUPPLY RADIATING SURFACES.

In addition to the foregoing it seems wise to reprint here an article by Earnest T. Child that appeared under the above heading in *The Metal Worker* of Aug. 19, 1899.

The primary method of figuring the sizes required is to ascertain the volume of steam which will be condensed by the radiating surface. This being known, the size of pipe may be computed by assuming a velocity of flow, which will cause a loss of pressure not exceeding 12 inches water head per 100 feet of pipe; say, a velocity of 50 feet per second, which will give an approximate frictional resistance of 8 inches of water per 100 feet.

For instance, to supply 1000 square feet of radiating surface at 5 pounds gauge pressure:

Temperature of steam =	227 degrees F.
Temperature of air in room,	70 degrees F.
Difference, air and steam,	157 degrees F.

British thermal units radiated per square foot of surface as per experiments by Wm. J. Baldwin, J. H. Mills, and others average 275. This gives 275,000 British thermal units per hour. As each pound of steam at this pressure is capable of yielding 954 British thermal units, it will require 288 pounds of steam per hour. One cubic foot of steam at 5 pounds weighs 0.0511 pounds, so 288 pounds equal 5636 cubic feet per hour, or 1.56 cubic feet per second, which, flowing at the rate of 50 feet per second, will require an area of 0.0312 square foot, or 4.59 square inches. This would require a pipe 2.4 inches in diameter, and as the next higher commercial size is 2½ inches in diameter, this would be the size required.

This is a roundabout way, however, and various formulas have been evolved for figuring the pipe sizes.

Robert Briggs in his "Steam Heating" uses a formula which has been extensively followed either directly or in a modified form, in which d = diameter of pipe; Q = 9.2 cubic feet of steam

per 100 square feet radiating surface; $l =$ length of main in feet; and $h =$ head of steam to produce the flow. From this

$$d = 0.5374 \sqrt[5]{\frac{Q^2 l}{h}}$$

Frederic Tudor used a modified form of the same, in which $C =$ volume of steam per minute $= 9.2 \times$ total radiating surface $\div 100$; $L =$ length of main in yards; and $H =$ head in inches of water lost [loss in] pressure; $d =$ diameter of pipe. From which

$$d = \sqrt[5]{\frac{C^2 \times L}{H}} \div 3.7$$

These two formulas, when figured on a basis of 6 inches lost [loss in] pressure in a 100-foot run, agree very closely.

A rule given by Wm. J. Baldwin in his "Steam Heating for Buildings," and also published by Geo. H. Babcock, states that diameter of main in inches should equal one-tenth of the square root of the total radiating surface, mains included. This rule, when compared with the two previous ones, provides for a much more ample pipe, and on systems of over 4000 square feet it would be safe to use on mains as long as 600 feet, though it does not primarily take in the element of distance at all. Even for smaller systems it gives a relatively large diameter, and for 10,000 square feet of surface it gives a diameter fully 35 per cent. larger than the best accepted practice, which means an area which is nearly doubled.

A. R. Wolff, in his "Addendum to Steam Heating" by Briggs, gives the following: "For determining the cross section area of pipes (in square inches) for steam mains and returns, it will be ample to allow a constant of 0.375 square inch in coils and radiators, 0.375 square inch when exhaust steam is used, 0.19 square inch when live steam is used and 0.09 square inch for the return for each 100 square feet of heating surface. If the cross sectional areas thus obtained are each multiplied by one and three-elevenths, and the square root extracted from each product, the respective figures will represent the proper diameters in inches of the several steam pipes referred to. The steam mains should never be less

than 1½ inches in diameter, nor the returns less than ¾ inch in diameter."

This rule does not take into account the relative decrease in friction in pipes of larger diameters, and while in systems under 2500 square feet it follows the Briggs and Tudor formulas very closely, on larger areas it goes up more rapidly on account of the fact that the area of the main is directly proportional to the area of the radiating surface. This formula is safe to use for mains under 200 feet in length, but if this be exceeded the area should be proportionally increased.

Prof. R. C. Carpenter, in his "Treatise on Heating and Ventilating Buildings," uses Briggs' formula, with the exception that for a frictional resistance of 6 inches water column he uses the value of 318.6 for *h* instead of 477.8, which gives a 50 per cent. larger area of main; but as this table is figured for a single pipe system, 50 per cent. larger areas will, of course, be necessary. In figuring for a separate return he uses the Briggs formula without change. His rule for bends and obstructions is as follows: "Right angle ells add 40 diameters; globe valve, 60 diameters; entrance tee, 60 diameters. For other resistances and steam pressures multiply the diameters by the following factors:

Water level above boiler	2 inches.	12 inches.	18 inches.
Multiply by	1.25	0.88	0.80
Steam pressure above atmosphere	0 pound.	3 pounds.	30 pounds.
Multiply by	1.03	1.02	0.97

For indirect heating with separate return use the result as obtained."

The result of the above formula when plotted gives diameters about ½ inch larger than A. R. Wolff's rule up to 5000 square feet, and at about 7600 square feet the lines cross (see chart). On the other hand, when the "double pipe system" line is plotted, it follows the Wolff and Tudor curves quite closely up to 4000 square feet, and at 7000 square feet it drops below them all.

The results of all the above formulas have been plotted on a chart presented herewith, and it will be seen that for practical work as well as handiness in figuring and ease in remembering a simple formula, A. R. Wolff's rule of 0.375 square inch per 100 square feet of surface is best adapted to ordinary conditions of low pressure heating.

A line for sizes of returns has also been plotted, based on 0.1 square inch for each 100 square feet of surface. This gives an area about 11 per cent. larger than that used by Mr. Wolff; but

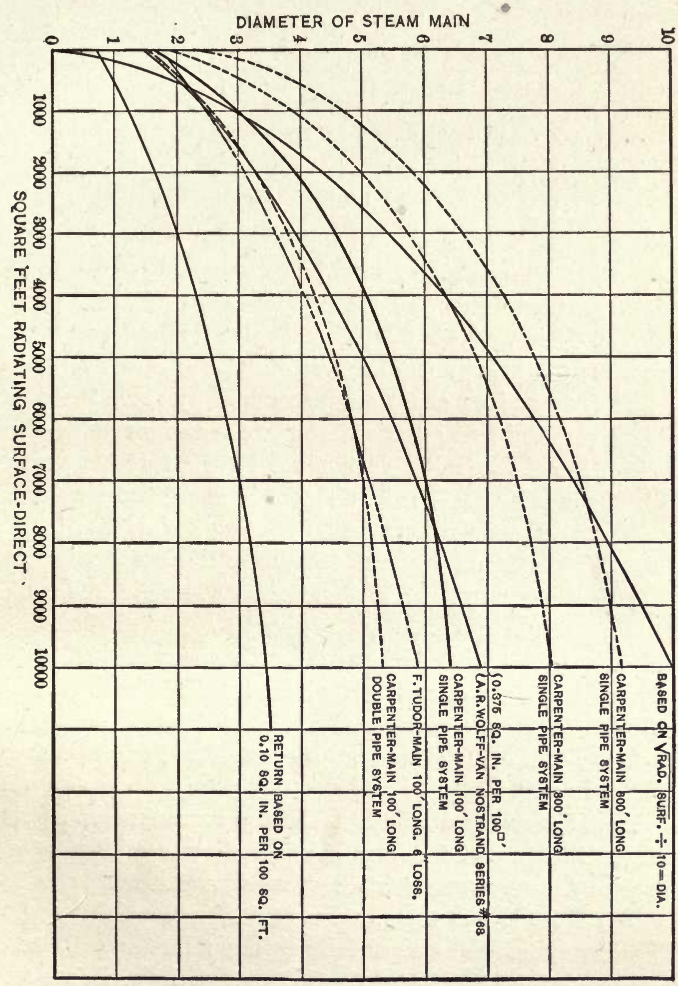

CHART 7.

Showing Size of Steam Mains According to Different Rules.

it will be noticed that the results are very nearly in line with the best practice.

Various rules for pipe sizes may be found, all of which vary in a greater or less degree, and seem to have been arrived at in a

more or less roundabout way, or by some rule of thumb. To illustrate the range covered by these rules a comparative table is given herewith, which may prove interesting:

TABLE XXXVII.

DIAMETER OF PIPE AND NUMBER OF SQUARE FEET SUPPLIED.

Name.	1-inch.	1¼-inch.	1½-inch.	2-inch.	2½-inch.	3-inch.	3½-inch.
Billings	..	..	225	450	700	1,200	1,500
Tudor	40	80	160	320	640	1,280	....
Nason	..	125	200	500	1,000	1,500	2,500
Willett	40	70	110	220	360	560	810
Wolff	60	120	200	480	880	1,500	....

Billings, in his " Ventilating and Heating," states that the only objection to having steam mains large is increased first cost, but this is a poor argument for an engineer to set forth, as it is his business to design a system which will give the best and most economical results at a moderate first cost. He also overlooks the fact that larger pipes cause a greater loss by radiation.

The sizes used by Frederic Tudor are for connections to radiators only, the mains being determined by the formula previously stated. This is a simple rule, and has been proven very satisfactory. He allows 40 square feet radiation for each 1-inch pipe, and doubles the figure for each successive size up to 3 inches.

The sizes given by J. R. Willett are very large indeed compared with other authorities, and are not to be recommended.

The sizes used by the Nason Mfg. Company and A. R. Wolff, as given on pages 540 and 541 of Kent's Handbook, are almost identical. Of these five cases the rule used by Tudor appeals to the writer [E. T. Child] as being the simplest and most practical, in that it is easily applied and not easily forgotten; and while it gives sizes which are ample to fulfill the requirements it does not overdo the matter. Of course single radiators seldom aggregate more than 300 square feet on direct work, and it will be noted that up to this size there is very little variation in the sizes used by all.

The formulas given in the foregoing relate entirely to direct radiation. It has been found that indirect stacks condense from 50 to 100 per cent. more steam than direct, depending upon the velocity of the air passing over them, other conditions being the same. R. C. Carpenter states in his " Heating and Ventilating Buildings ":

"The indirect heating surfaces require about twice as much heat as the same quantity of direct radiating surface, and hence, for same resistance in the pipe, the area should be twice that required in direct heating. It will usually be sufficiently accurate to use a pipe the diameter of which is 1.4 times greater than that for direct heating." But he makes a statement earlier in the book that for indirect heating with separate return an area 50 per cent. larger than that used for direct heating will be sufficient. To cover all contingencies, however, it will be safe to figure an area twice as large as for direct, and the same rule, of course, applies to the return.

Indirect heaters, when used in connection with a fan, condense even more steam than when operated under natural draft, on account of the greater velocity bringing more air into contact with the radiating surface in a given time. The quantity of steam which will be condensed in them under these conditions is, however, a decidedly variable quantity. Suppose, for instance, that the air entering the heater is at 0 degrees F., then the condensation will be 36 per cent. greater than it would if air were returned from the building at 60 degrees F.; velocity being constant and steam pressure 5 pounds gauge. If the velocity of air passing through the heater changes from 750 to 1500 feet per minute there will be a further increase of at least 30 per cent.; making a total variation of about 77 per cent. in the amount of steam condensed. J. H. Mills states that 1000 cubic feet of air passing over each square foot of surface will cause it to condense from 900 to 1300 British thermal units.

LOW PRESSURE HEATING MAINS.

The following extracts from an article by C. E., which appeared in *The Metal Worker* of June 25, 1904, are reprinted here as adding something to the general fund of information on this subject:

"Gradually a set of rules for accurately determining the size of steam mains is being evolved. One of the earliest of these, and one of the most extensively used, states that a square inch of free cross sectional area in a steam pipe will supply 100 square feet of radiating surface. This rule is qualified by its originator in many different ways, so much so that he is reputed to have said that if

a pipe proved too small to double its size. This rule totally neglects the varying amount of frictional resistance between large and small pipes. It is rather absurd, of course, to assume that the carrying capacity of an 8-inch pipe can be computed by the same rule as a 1¼-inch pipe."

There are numerous other rules which have appeared in the more recent scientific books, all of which are helpful in their way, but none of which is in very general use among engineers, owing to the fact that they cannot be applied to pipes of all sizes.

Probably one of the safest rules in calculating the size of a steam heating main is that in common use among engine builders —that is, basing the size of the pipe to give an arbitrary velocity of steam flowing through. In high pressure work the safe velocities are well known; but in low pressure work this is not so, as there are only a few offices in which this method of calculating sizes has been experimentally reduced to a comparatively exact science and in which the safe velocities for various sizes of pipes used for different purposes are definitely known.

The basis, of course, for any such rule must be the amount of steam condensed by a direct radiator of the usual type working under normal conditions with the outside temparature at zero. After an exhaustive series of experiments it has been determined that this will amount to approximately 0.3 pound of steam condensed per hour per square foot of radiating surface. This amount, 0.3 pound, is based on using steam at zero pressure; but as the ordinary steam heating plant is designed to operate at 1 to 10 pounds pressure, the difference in the amount of condensation at pressures within that range, although considerable, would not be enough to overload liberally designed piping.

Given the square feet of heating surface, the cubic feet of 1 pound of steam and the safe velocity, it is an easy matter to determine the size of the piping. The only difficult part is to determine what is the safe velocity for a given condition. No set of calculations, no matter how elaborate, will give this; nor can one fall back on the experience of the steam fitter, as he hasn't the slightest idea how fast the steam is going.

The best sources of information available indicate that the following velocities are safe. They are based on extensive experi-

ments and observations among old buildings in which the piping is very small: A velocity of 80 feet per second is perfectly safe in mains 2 to 3½ inches, inclusive. On mains of these sizes the frictional resistance is rather high, so that the velocity used is low. Still, even at that, a 3-inch main will supply 1800 square feet of direct radiation. According to the old rule of a square inch of area to 100 square feet of surface, the same pipe would supply only about 750 square feet—rather a wide variation between two rules; yet the former has been demonstrated time and again to be perfectly true.

For 1¼ and 1½-inch mains the safe velocity is hardly more than 50 feet per second; but, as a matter of practice, these sizes are rarely used with any but an arbitrary amount of radiation, depending on local conditions. At 50 feet velocity a 1½-inch pipe will supply 300 square feet of radiation. A velocity of 90 feet is low enough for 4 to 6 inch pipe, inclusive. On this basis a 5-inch main will supply 5700 square feet. This is probably considerably more than current practice among steam fitters allows. On pipes larger than 6 inches a velocity ranging from 95 to 100 feet per second is considered good practice. An 8-inch pipe at 100 feet velocity will carry about 15,000 square feet of direct radiation, and a 12-inch pipe about 35,000 square feet.

It is presumed, of course, in giving the above figures that the pipes are insulated with a fair make of covering and that they are reasonably dripped.

An elaborate system of drips is not essential, but the importance of a reasonable dripping cannot be overestimated. A main cannot be expected to carry its maximum amount of surface if in addition it must carry the condensation from a long system of mains. Furthermore, it is necessary that the drips be made in a way that will avoid any churning of water in the fittings at the drips. There is nothing so fatal to the capacity of a main as the churning and splashing caused by badly made drips and by wrong pitch.

For continuous circuit main work, so largely used nowadays, especially in the smaller class of buildings, it is necessary to provide carrying capacity in the mains for the entire amount of condensation as well as the steam, although it may be urged that as the

amount of water increases the amount of steam decreases. Still it is usual to make large allowance for the water in this class of work, using a velocity of about 60 feet for the smaller size mains and 70 feet for the larger sizes. On this basis a 5-inch continuous circuit main will supply about 4000 square feet of radiation.

The proper proportioning of the risers in a heating plant is probably the most difficult part. It is of course fatal to the entire apparatus to get them too small; and, at the same time, structural conditions usually, and the wishes of the architect or owner, necessitate making them as small as possible. A low velocity must be used on account of the reverse flow of water; much more serious in one-pipe work than in two-pipe. A velocity of 40 feet per second is perfectly safe on one-pipe risers and 50 feet for two-pipe risers. On this basis a 2½-inch one-pipe riser will supply 600 square feet of radiation and 2½-inch two-pipe riser about 750 square feet. These figures may seem excessive, but they are constantly in use and give excellent results. [These are greatly in excess of the capacities given in Author's Tables XXXIII and XXXIV.]

No set velocities for radiator connections can be given, as these are determined arbitrarily by good practice, it being necessary to make allowance for many other things besides the amount of steam a connection will normally carry. The sizes are well known and will not be repeated here.

It is essential, in designing any steam heating apparatus, to provide for the very heavy demand for steam when the plant is put in operation in the morning. The effect of this, of course, is to increase the velocities, which effect is most troublesome in the risers and radiator connections. The velocities as given above are sufficiently low to provide for this, so that no further allowance need be made. It will be noticed that the risers will be far larger than the mains in proportion to the amount of steam they carry. Radiator connections in good practice are made larger than any possible demand for steam would necessitate.

SIZES OF MAIN STEAM PIPE CONNECTIONS WITH BOILERS.

Suppose a boiler is supplying steam to an engine cutting off at, say, one-third of the stroke—that is, admitting steam about one-third of the time? Assuming a maximum velocity in the supply

pipe of 6000 feet per minute, if steam is passing through the same only one-third of the time, the average velocity will be 2000 feet per minute. Basing the size of main steam connections with boilers on this velocity gives the following size pipes when the steam pressure is 80 pounds. The pipe sizes for higher pressures would, of course, be smaller if computed in the same manner, but it is advisable to use as large pipes as those stated in the table, which conform pretty closely with present boiler practice:

TABLE XXXVIII.

SIZE OF MAIN STEAM PIPES FOR BOILERS OF HORSE-POWER STATED.—STEAM PRESSURE ASSUMED TO BE 80 POUNDS BY GAUGE; AVERAGE VELOCITY IN PIPE, 2000 FEET PER MINUTE.

Boiler horse-power.	Pipe area. Square feet.	Size of pipe corresponding. Inches.
50	0.057	3
62½	0.071	3½
75	0.085	4
100	0.114	4½
125	0.142	5
150	0.171	6
200	0.228	7
250	0.285	8
300	0.342	8

NOTE.—Four and one-half inch pipes and valves being an odd size, it is advisable to use 5-inch instead. When globe valves are used in boiler connections, it is well to make the pipes one size larger than when straightway gate valves are used, to compensate for the increased resistance.

SIZES OF STEAM AND EXHAUST PIPES FOR ENGINES.

The steam ports and supply pipes to engines are commonly proportioned on a basis of a maximum velocity flow of 6000 feet per minute. A simple automatic or throttling engine running on, say, 80 pounds steam pressure and taking 30 pounds of steam per horse-power per hour would require about 137 cubic feet of steam at the pressure stated for each horse-power per hour. The admission of steam is cut off anywhere from one-quarter to three-quarter stroke; seldom over one-half stroke, unless the engine is very much overloaded. If we assume a cut-off of four-tenths of the stroke as a fair basis on which to compute the horse-power for pipes of different sizes we have under these conditions the capacities stated in the following table.

TABLE XXXIX.

SIZES OF SUPPLY PIPES FOR STEAM ENGINES.

Nominal diameter of pipe in inches.	Engine horse-power supplied.
2	24
2½	35
3	54
3½	72
4	92
4½	117
5	145
6	210
7	283
8	364

Engines exhaust during almost the entire stroke—say 95 per cent. as a fair average. On this basis, assuming 1 pound back pressure, 30 pounds steam per horse-power per hour and a maximum velocity through the exhaust pipe of 5000 feet per minute, the appropriate horse-power for exhaust pipes of given sizes has been computed and is stated in the following table: [4000 feet velocity is not an uncommon velocity to assume.]

TABLE XL.

SIZES OF EXHAUST PIPES FOR STEAM ENGINES.

Nominal diameter of exhaust pipe in inches.	Engine horse-power.
2½	20
3	30
3½	40
4	50
4½	63
5	80
6	115
7	153
8	200
10	312

A comparison of Tables XXXIX and XL shows the size exhaust pipe for a given horse-power to be one size larger than the steam pipe, which accords very well with the general practice of engine builders. Some engine builders make their steam and exhaust connections abnormally large to provide for cases where the pipe lines are long. The foregoing tables give permissible sizes that may be used in proportioning the piping in office and other buildings having individual or isolated mechanical plants.

EFFECT OF BACK PRESSURE ON SIMPLE AUTOMATIC ENGINES.

With a simple automatic engine carrying a back pressure of 5 pounds the loss in power due to back pressure will be as follows: Take, for example, a high speed engine commonly used to drive a direct connected dynamo. With 90 pounds initial gauge pressure, equal to about 105 pounds absolute pressure, and steam cut off at one-quarter stroke, the average pressure per square inch on the pushing side of the piston throughout the stroke will be about 63 pounds.

From this must be deducted the atmospheric pressure, equal to 15 pounds per square inch, or say 16 pounds, to allow for the resistance of the exhaust pipe and elbows. The mean effective pressure, equal to the average pressure on the pushing side of the piston minus that on the exhausting side is $63 - 16 = 47$ pounds. Now with 5 pounds back pressure, or a total of 20 pounds above a vacuum, the steam pressure on the pushing side remaining the same, the mean effective pressure will be $63 - 20 = 43$ pounds. The horse-power will be in proportion to the mean effective pressures computed above; that is, with 5 pounds back pressure the engine will have only 43-47ths or 91½ per cent. of the horse-power it has when exhausting freely to the atmosphere. In other words the loss in power due to the back pressure would be nearly 9 per cent.

EFFECT OF BACK PRESSURE ON COMPOUND ENGINES.

The effect of back pressure is a more serious matter in the case of compound engines than with simple ones, since it acts on the relatively large area of the low pressure piston. To show to what extent the engine horse-power is reduced by a 5-pound back pressure on a compound engine, take for example an engine with a 16-inch high pressure cylinder, a 24-inch low pressure cylinder and a 16-inch stroke. A 5-pound back pressure exerted over the large area of the low pressure piston would with a piston speed of 600 feet per minute amount to 452 (square inches) $\times$ 5 (pounds) $\times$ 600 (feet) $\div$ 33,000 (foot pounds per horse-power) $=$ 41 horse-power. Such an engine with 125 pounds gauge pressure when run non-condensing is rated to develop about 225 horse-power, hence an increase in the back pressure of 5 pounds decreases the effective output of the engine about one-fifth or 20 per cent.

COUNTERACTING BACK PRESSURE BY INCREASED BOILER PRESSURE.

With a back pressure exhaust heating system either larger engines must be used to secure a given horse-power or a higher boiler pressure must be carried. If the latter is done considerably more than the 5 pounds back pressure commonly allowed on the engine must be added to the boiler pressure, since the back pressure is maintained throughout the stroke of the engine, but the boiler pressure is cut off at one-quarter, one-third, or some other point of the stroke, as the case may be. To counteract 5 pounds added to the back pressure of an engine cutting off at one-quarter stroke about 8 pounds must be added to the boiler pressure. A few pounds added in this way is not a serious matter so far as fuel consumption is concerned, since the total heat necessary to make steam increases very slowly with an increase in pressure and not at all in proportion to the pressure increase. With ordinary tubular boilers, however, the allowable pressure that may be carried is cut down from time to time by the insurance companies, so that if 10 pounds more pressure must be carried, for example, to overcome a certain back pressure than would otherwise be necessary, the boiler must be condemned so much the sooner.

STEAM HEATING IN CONNECTION WITH CONDENSING ENGINES.

In the case of plants having condensing engines, either simple or compound, the question arises whether it is better economy to run the engines noncondensing part of the time and heat with the exhaust steam, or to always run them condensing and heat with live steam. Which is the better policy depends chiefly on the amount of steam required for heating in comparison with the total exhaust from the engines. If the amount is very small manifestly it would be better to run condensing and secure the marked saving in steam and supply the heating system with live steam through a reducing valve. When there are several engines it is well to have the exhaust pipes connect with a header with cut-out valves between the engines, one end of the header connecting with the condenser and the other with the line leading to the heating system. Then one, two or more engines may be run condensing and the others exhaust to the heating system.

As to the economy: Suppose a compound condensing engine will develop an indicated horse-power with the consumption of

16 pounds of steam per horse-power per hour, and will require 23 pounds of steam to develop a horse-power when running noncondensing, a difference of 7 pounds. Under the conditions stated a 300 horse-power engine would consume when running noncondensing $300 \times 23 = 6900$ pounds per hour. If the engine were run condensing it would consume $300 \times 16 = 4800$ pounds. In the case assumed whenever more than $6900 - 4800 = 2100$ pounds of steam per hour are required by the heating system it will be cheaper to run noncondensing.

Suppose for example that 3600 pounds of steam are necessary to supply the heating system for one hour. If the engine is run condensing 4800 pounds of exhaust steam will be condensed and 3600 pounds of live steam must be supplied, a total of 8400 pounds in one hour, whereas if the engine were run noncondensing 6900 pounds of exhaust steam would be secured, of which 3600 would be used for heating, the rest escaping through the exhaust head, except that utilized in heating the feed-water.

With the steam consumption assumed, whenever more than 7 pounds of steam may be utilized in the heating system to each horse-power developed by the engine it would be better economy to run noncondensing. When less than 7 pounds is needed the engine should be run condensing. For example, suppose steam is required by the heating system at the rate of 4 pounds to each horse-power developed by the engine, one horse-power condensing will take 16 pounds of steam, which, plus the 4 pounds of live steam supplied to the heating system, amount to 20 pounds per engine horse-power, whereas if the engine were run noncondensing 23 pounds would be consumed. Against this method of heating must be charged the larger size engine required to produce a given horse-power when running noncondensing.

In the case of a Corliss simple noncondensing engine taking, say, 26 pounds of steam per horse-power per hour and 21 pounds when condensing, it will be found cheaper to run noncondensing whenever the heating demands more than 5 pounds of steam to each horse-power developed by the engine; in other words, whenever the steam for heating exceeds more than about one-fourth that for power it will be better economy to run noncondensing, and when less than that amount to run condensing.

CHAPTER IX.

MODIFIED SYSTEMS OF STEAM HEATING.

Under the above heading will be described the essential features of vacuum, vapor and fractional valve systems of steam heating. These methods have to a great extent taken the place of the old-fashioned plain one-pipe and two-pipe systems with air valves.

While differing in details these modified systems may be divided in general classes which will be described to an extent that will make clear their essential features; readers in search of more specific information are referred to manufacturers' catalogs.

TWO-PIPE VACUUM SYSTEMS.

In the Webster system, the best known two-pipe method of vacuum heating, the steam supply to the radiators is controlled by a hand valve as in the ordinary two-pipe system. At the return end of each radiator is placed an automatic water and air relief valve which permits the escape of air and water and prevents the escape of steam. Since air is heavier than saturated steam, in the ratio of 1 to 5/8 at atmospheric pressure, the location of this valve at the lower end of the radiator opposite the steam inlet is stated to be the most effective one possible. Air valves are not required with this system. Typical radiator connections are shown in Fig. 36. It is essential that each unit of radiation and each drip point in supply lines (unless a special system of drips be provided) be equipped with one of these automatic return valves, otherwise any unit or drip point without one would permit steam to pass into the returns and destroy the vacuum which it is the function of the pump to maintain. By means of this suction a continuous removal of condensation and air from the heating system is secured.

The water and air drawn from the system is discharged by the vacuum pump to an air-separating chamber. When a closed

feed-water heater is used the vacuum pump discharges to an open receiver from which the water is pumped through the heater to the boiler.

The vacuum pump exerts a suction on the return ranging as a rule from 4 to 12 inches mercury column, according to the length and size of the pipes. With this system high-pressure returns should not as a rule be connected with the returns leading to the vacuum pump, since the high temperature of the condensation causes a portion of the water to vaporize in the returns

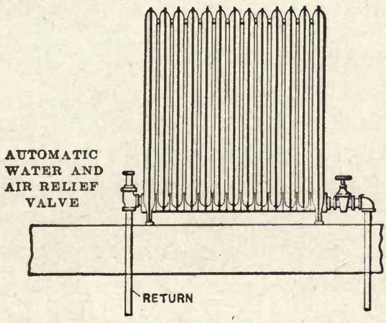

Fig. 36.—Typical Radiator Connections.

and interferes with the maintenance of the vacuum. The exhaust from the pump is utilized in the heating system.

In buildings heated by this system it is possible to supply an amount of steam less than that required to completely fill the system at atmospheric pressure, hence a saving may be made by heating at night or during mild weather with steam at a pressure below that of the atmosphere.

It is often necessary to locate some radiating surface at a point lower than the main return. With the vacuum system the condensation may be raised several feet above the level of the radiator to be drained, by reason of the suction in the returns.

The back pressure and pressure reducing valves need not be set to produce initial pressure in the heating mains in excess of that required to supply the most remote unit of radiation with steam at atmospheric pressure.

In the case of high pressure plants, two reducing valves, set

tandem, are sometimes installed, the first to reduce from boiler pressure down to 15 or 20 pounds or thereabouts, the latter to reduce to atmospheric pressure or to a few ounces above the pressure of the atmosphere.

Among the advantages claimed for two-pipe vacuum systems are:

1. Absence of back pressure on motive engines when exhaust steam is utilized.

2. A perfect drainage of supply pipe systems preliminary to an equally perfect drainage of radiating surface without the loss of steam.

3. A continuous automatic drainage of condensation and the prevention of any accumulations of water.

4. A positive and consequently effective steam circulation.

5. Perfect control of circulation with power to vary it at will.

6. Removal of air and gases from heating surfaces and feed water.

7. Power to independently modulate temperature in any part of the heating surface.

8. The return of condensation from points somewhat below the line of drip or drainage mains when necessary.

9. Smaller pipes may be used than with the ordinary low pressure two-pipe system.

It is pointed out by the manufacturers that the positive removal of air from the radiators is alone a great advantage, since automatic air valves seldom properly perform the function for which they were designed, and unless air lines lead from them to some suitable point of discharge the ill-smelling air from the radiators is discharged into occupied rooms.

Water hammer, due to ignorance or carelessness in operating radiator valves, is entirely overcome by the use of the two-pipe vacuum system. The supply valve is the only one that requires any attention, the return being automatic. The supply of steam may be throttled down at will.

The steam pressure in the radiators is not reduced by the vacuum maintained on the return, but depends solely on the amount of steam admitted to the radiators. Indirectly the vacuum on the return affects the steam pressure, since no pressure

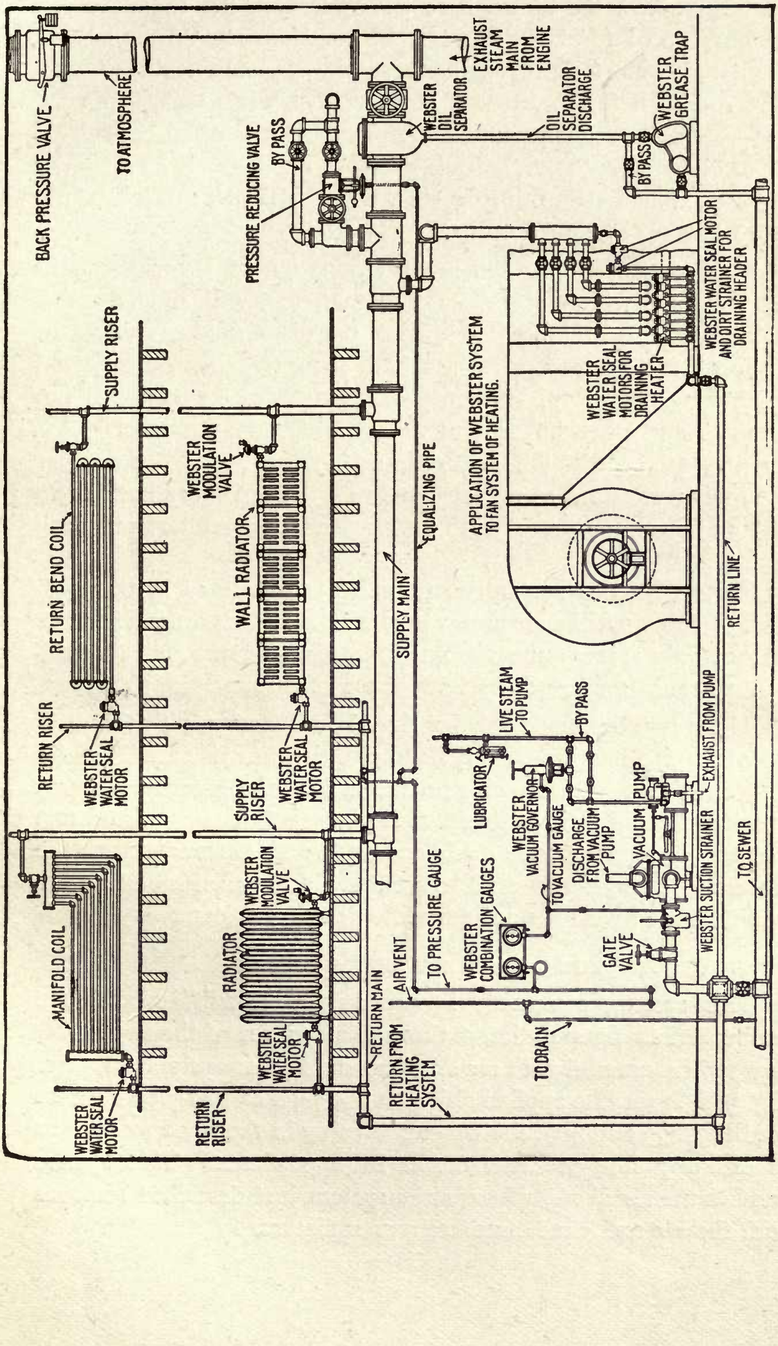

Fig. 37.—General Arrangement of Two Pipe Vacuum System.

whatever above the atmosphere is required in the radiators for the purpose of forcing the water of condensation through them and the air out of them. In the case of old plants having insufficient radiation for the most severe weather, when using the very low pressures common with vacuum systems it is often better policy to carry a few pounds back pressure on the engines furnishing the exhaust steam during such weather than to overhaul the entire heating system.

This system secures the ready circulation of steam throughout buildings widely separated, and that, too, with only a slight back pressure on the engines. With the usual methods of steam heating it is necessary to carry a back pressure, even during mild weather, when the full efficiency of the radiating surfaces is not required, and when but few of the radiators of an extensive system may be needed. Under certain conditions it would be cheaper to supply live steam at reduced pressure than to carry back pressure on the engines in order to supply a small amount of radiating surface.

Since with this system no pressure is necessary in the radiators to force out the air and water, it follows that a drop in pressure of only a few ounces from the initial pressure will be sufficient to cause the necessary flow of steam through the pipes.

These may be made smaller than is customary with the ordinary two-pipe low pressure system, and the returns may be decidedly cut down in size owing to the action of the vacuum pump creating a rapid flow in them. See pipe sizes, pages 106 and 107. The supply pipes may be made one or two sizes smaller with the vacuum system, and the returns two to three sizes smaller than would be used with the ordinary low pressure system.

THE AIR-LINE VACUUM SYSTEM.

This system, commonly known as the "Paul," secures the removal of air from radiators through air valves of the expansible plug type connected with air lines leading to a steam ejector. See Fig. 38. It may be applied either to one-pipe or two-pipe systems (see Figs. 39 and 40), the water returning in the same manner as in ordinary low pressure steam heating plants. This system handles the air alone, whereas the system just described removes both the air and condensation from radiators.

That air is the most serious hindrance to the proper operation of a steam heating plant is a well-known fact. To attempt to get rid of it by forcing it through ordinary automatic air valves by steam pressure is a rather slow process, especially in the case of large coils or radiators. With a common low pressure system the air remains in the radiators until forced out by the steam. With the vacuum system the air may be removed from the radia-

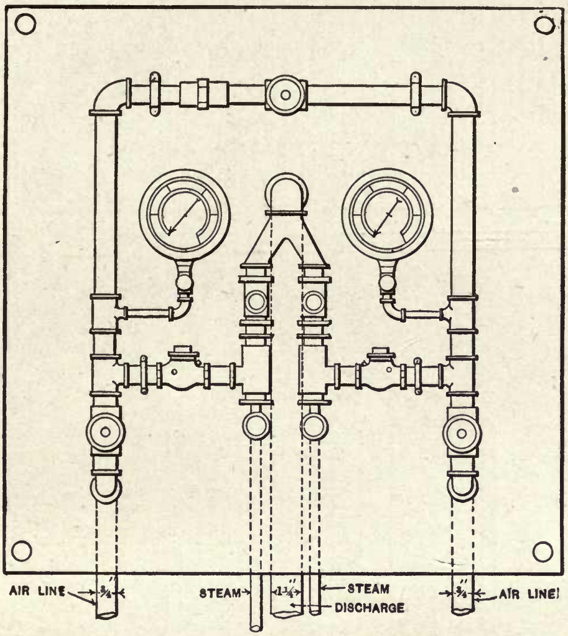

Fig. 38.—Front Elevation of Exhausting Apparatus.

tors by starting the ejector before steam is turned on the system. The radiators then become quickly filled with, and remain full of, steam, since the air is automatically removed as rapidly as it accumulates.

ABSENCE OF BACK PRESSURE.

One of the chief advantages of this system over ordinary low pressure heating is the removal of back pressure from the engines and pumps. By exhausting the air from the radiators by means

of the steam ejector they become practically condensers, the engines exhausting into them. In manufacturing plants where the

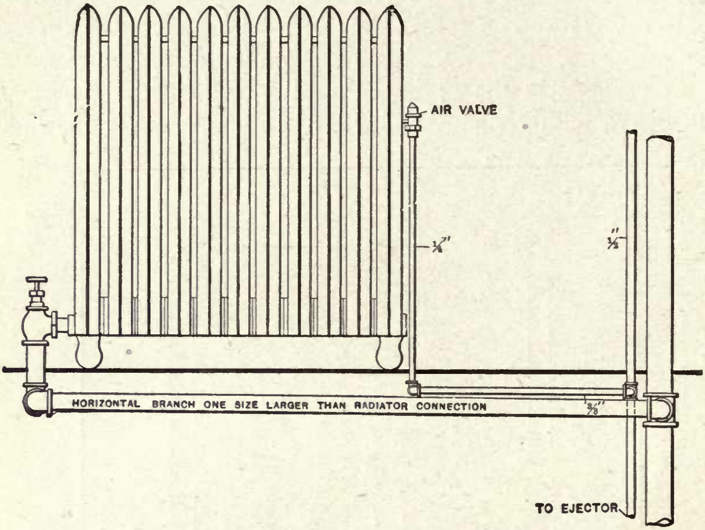

Fig. 39.—Connections for One-Pipe System.

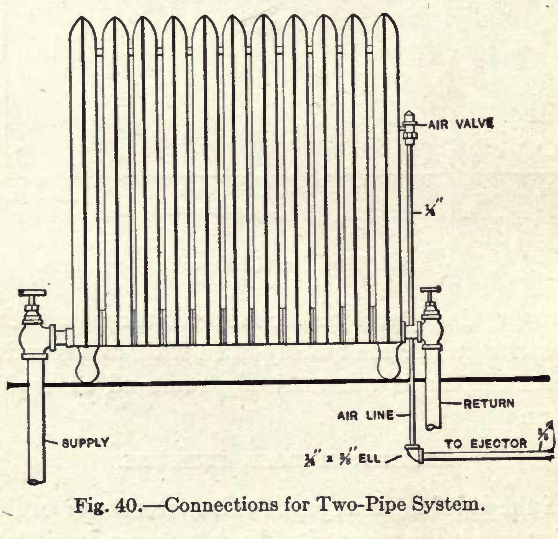

Fig. 40.—Connections for Two-Pipe System.

power requirements may be in excess of those for heating the importance of the elimination of back pressure is apparent.

STEAM TO OPERATE THE EJECTOR.

As to the amount of live steam required to operate the ejector: To compute the volume of steam escaping from an orifice to the atmosphere, allow about 900 feet velocity per second and multiply by the area of the opening expressed in the decimal part of a square foot. As to the amount of steam required to operate the ejector, A. B. Franklin, in a paper on exhaust steam heating, read before the Master Steam and Hot Water Fitters' Association of the United States, June 7, 1893, states that in ten hours' run, with a fan system of heating having heaters containing an aggregate of 24,150 linear feet of 1¼-inch pipe, supplied by a 6-inch main, the ejector discharging to a condenser used 300 pounds of steam in that length of time. A test made at the Ohio State University showed the total weight of water returned from the radiators to be 8,160 pounds and the steam used by the exhauster or ejector to be 432 pounds.

Claims for this system are:

1. A positive and uniform circulation of steam without pressure above that of the atmosphere.

2. Utilizing the heat of steam at low temperatures, thereby gaining great economy.

3. Warming without impairing the quality of the air in the rooms.

4. The independent and automatic removal of the air and water of condensation from the heating apparatus.

5. A sealed system; no leakage, no smell or dripping from air valves.

6. All heating surface held in the best condition to operate promptly when desired, and all parts of the surface rendered uniformly efficient when steam is turned on.

7. Exhaust steam utilized without back pressure at engine or pumps.

8. The water of condensation returned quickly and economically at highest temperatures.

9. Less steam used, less coal burned, to heat a given space.

HEATING WITH RADIATORS AT A RELATIVELY LOW TEMPERATURE.

Professor Kinealy, reporting on some tests to show the effect of the relatively low temperatures secured by the use of a vacuum system, makes the following statement: "The radiator at high temperature probably kept the air at the top of the room, when the temperature about 5 feet from the floor was 70 degrees, at a much higher temperature than it was kept when the temperature in the radiator was low. The higher the temperature of the air at the ceiling of the room the greater will be the average temperature of the air in contact with the cooling windows and walls of the building, and therefore for a given outside temperature the greater will be the difference between the average temperature of the air inside and that of the air outside, and hence the greater will be the amount of heat transmitted through the cooling walls and windows per hour. As the occupants of heated rooms live in the air which is within 6 feet of the floor, that system of heating must undoubtedly be the best and the most economical which will maintain the desired temperature of the room nearly uniform from the floor to 5 feet above it, with a low temperature in the upper part of the room, and this is, I think, done by radiators supplied with steam at low temperatures." (See "Heating with Steam at or Below Atmospheric Pressure," by J. H. Kinealy, in *The Metal Worker, Plumber and Steam Fitter,* July 29, 1899).

THE POSITIVE DIFFERENTIAL SYSTEM OF STEAM CIRCULATION.

In this system a controlling valve is placed at the foot of return risers as indicated in Fig. 41. These valves are designed to maintain any desired difference in pressure between the supply and return risers. By maintaining this constant pressure difference it is claimed that a special type of swing check valve may be used at the return end of each radiator. A small opening is provided for the removal of air when this valve is closed. When open, both air and water pass through it.

Fig. 41 illustrates the application of this system to coils. The main supply riser is drained through a siphon loop to the return. It is claimed, since all return risers may be kept in the same con-

dition by means of the controlling valve that all air and condensation is drawn away from the radiating surfaces to the vacuum pump.

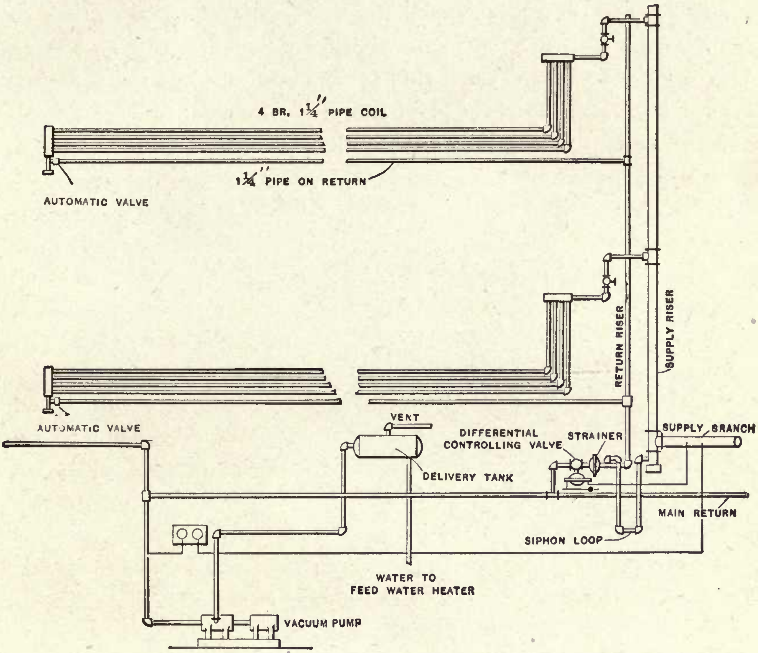

Fig. 41.—Application of the Positive Differential System to Radiating Coils.

FRACTIONAL VALVE SYSTEMS OF STEAM HEATING.

These systems of low pressure steam heating provide for the control of the heat emitted by radiators by regulating the admission of steam to them. A control valve is connected with the inlet of each radiator. These valves are capable of adjustment to admit enough steam to fill one-quarter, one-half, three-quarters or a fractional part of the radiator, a dial being provided to indicate the degree of opening.

At the return end of each radiator is placed a combined air valve and expansion or float trap. This valve is designed to allow water and air to escape from the radiator, but to prevent the escape of steam. Valves of the thermostatic type are oper-

ated by a liquid sealed in a suitable chamber. When steam comes in contact with it the liquid is vaporized and creates a pressure sufficient to force the valve disk or spindle against the seat. When the liquid cools, the valve opens and permits water and air to pass to the return pipe.

Fig. 42 shows a radiator equipped with supply and return valves as described. With this system radiators of the hot water type are preferable to those of the ordinary steam pattern, as the control valve may be more conveniently located and because the

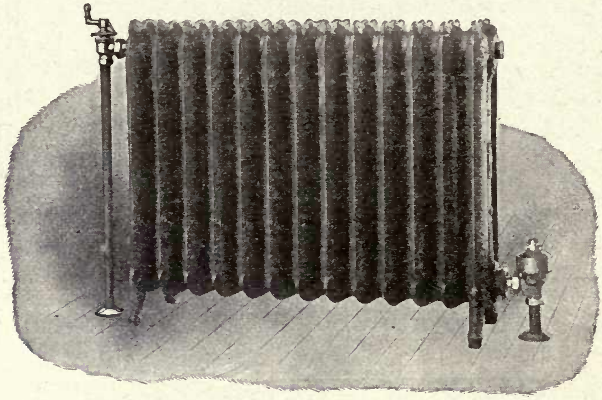

Fig. 42.—Radiator Equipped with Fractional Supply Valve and Automatic Return Valve.

circulation in the radiator is said to be somewhat better than with steam radiators.

When a control valve is partially closed the steam is condensed in the upper portion of the radiator, the lower portion is cold and becomes filled with air that backs up through the return valve or trap, the return risers being open to the atmosphere and free from pressure.

When the water of condensation is returned to a low pressure boiler it must be permitted to back up the returns above the water line in the boiler sufficiently to overcome the boiler pressure acting on the return where it connects with the boiler. The lowest radiators and drip points must therefore be well above the water line of the boiler. Since 1 pound pressure is equal to about 2.3 feet of water head, the maximum pressure that may be carried without flooding the radiators is limited by the available head, unless

special apparatus be provided. The use of a return tank and automatically controlled pump is recommended for large jobs. When the condensation is returned to a tank, as in large buildings, the tank must be vented to the atmosphere.

The piping of a fractional valve system is practically the same as a regular two-pipe system except that the returns must be open to the atmosphere. The main returns are run wet or dry, as best suits the conditions.

Advantages claimed for these systems are:

1. Positive circulation, due to absence of pressure at the return end of the radiators.

2. Quietness of operation.

3. Control of each unit of radiation independent of others.

4. Absence of separate air valves and lines, these being combined with the return carrying the water of condensation.

5. Convenience in operation, there being but one valve to manipulate.

6. Saving in fuel, due to absence of overheating in rooms, the heating being more easily controlled than with ordinary steam heating systems.

7. The quick heating of radiators, due to the rapid expulsion of air, there being no steam pressure in the returns to be overcome.

8. The drop in pressure between the supply and return lines being greater than in the ordinary two-pipe system, somewhat smaller pipes may be used if necessary.

GRAVITY RETURN VACUUM SYSTEMS.

In one of these vacuum systems air valves of the expansible plug type are attached to the radiators, and air lines are joined and led to the mercury seal shown in Fig. 44, and at A in Fig. 43, which shows a typical lay out of a one-pipe system. This method of piping is considered preferable, not only because of the greater convenience of having but one valve on each radiator, but because the fewer the valves the less the leakage through stuffing boxes, causing the loss of vacuum.

With this system special care must be exercised in packing radiator valves to prevent air leaking into the system and destroy-

ing the vacuum. It is claimed, however, that since the valves are used much less frequently than with low pressure systems, as

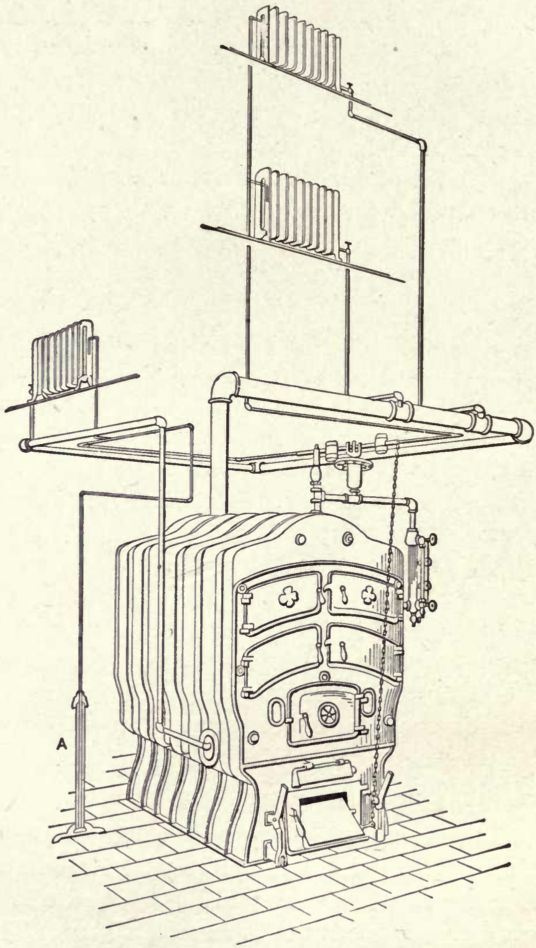

Fig. 43.—Trane One-Pipe Vacuum System.

tne temperature of the house is approximately controlled by varying the vacuum on the system, the stuffing boxes receive less wear, and if well packed give little trouble from leakage.

This system is so arranged that with a steam pressure of a

pound or two, the air in the radiators will be forced through the mercury and out of the system. The air valves prevent the escape of steam from the radiators. When the steam pressure is allowed to fall air is prevented from entering the system by the mercury column which rises in the pipe. The vacuum has no effect on the water line in the boiler, as the pressure on supply and return lines is the same. Of course, it will be necessary from time to time, even in mild weather, to get up sufficient pressure to expel the air from the system, as no job of piping can be made perfectly tight. Every precaution must be taken, however, to make the system as tight as possible, and all lines should be thoroughly tested with at least 30 pounds pressure, which should be carried for 24 hours without serious loss.

With this system it is recommended that gauge cocks be omitted from the water column, as they are sometimes the source of air leakage when the system is running under vacuum. The gauge glasses should be thoroughly packed; the stuffing boxes on radiator valves must be tightly packed. The same care must be taken to prevent pockets in the piping as with a regular low pressure system. The patentees recommend that the air pipe and fittings be made of galvanized iron to avoid trouble from stoppage by scale, etc.

Damper regulators of special design are used in connection with mercury seal systems, or thermostatic control may be applied, operating the boiler drafts from the thermostat located at a point that will represent as nearly as possible the average temperature of the house.

ADVANTAGES CLAIMED FOR THE MERCURY SEAL VACUUM SYSTEM.

The principal advantages claimed for this system of steam heating over ordinary ones are:

1. That it is as well adapted to mild weather as cold, whereas with a steam heating system a temperature of 212 degrees must be attained to secure any effect from the radiators.

2. That considerable saving in fuel may be effected in mild weather, due to the circulation of steam below atmospheric pressure, thus avoiding overheating, so common with low pressure

steam heating. In many sections in the northern part of this country the average outside temperature during the heating season is 35 to 40 degrees above zero.

Steam heating systems based on 70 degrees in zero weather are difficult to control with an outside temperature of, say, 40 degrees, when the loss of heat from a building is only about three-sevenths that in zero weather. The difference in temperature between the steam or vapor and the air in the room need be, under the stated conditions, only three-sevenths as much as in zero weather.

3. The mercury seal vacuum system when applied to a steam heating apparatus secures a wide range in the temperature at which the radiators may be kept to provide for different weather conditions.

The lack of this range of temperature in the ordinary low pressure steam system is the greatest drawback to its successful use in house heating. It is said to be practicable to maintain temperatures varying all the way from 170 to 230 degrees or more, which would permit the system to meet practically any outside weather conditions.

The following table shows the temperatures corresponding to different pressures:

TABLE XLI.

SHOWING STEAM PRESSURE AND VACUUM AND CORRESPONDING TEMPERATURE.

In. of mercury. Vacuum gauge.	Temperature. Fahr.	Gauge pressure. Lb. per sq. in.	Temperature. Fahr.
28	101.4	0.304	213.0
26	125.6	1.3	216.3
24	147.9	2.3	219.4
22	152.3	3.3	222.4
20	161.5	4.3	225.2
18	169.4	5.3	227.9
16	176.0	6.3	230.5
14	182.1	7.3	233.0
12	187.5	8.3	235.4
10	192.4	9.3	237.8
5	203.1	10.3	240.0
0	212.1		

The pressures are not given in even pounds. The 1.3 pound gauge pressure corresponds to 16 pounds absolute pressure, and so on.

COMPARISON WITH HOT WATER HEATING.

Advantages claimed for this system over hot water heating are:

1. Saving in cost of installation, as the pipes may be made smaller and smaller radiators may be used, owing to the higher temperatures carried in cold weather.

2. The ability to increase or decrease the temperature in the radiators more quickly, owing to the much smaller volume of water in the system.

3. The absence of danger of damage from leaks.

On the other hand, in weather, say, from 50 to 60 degrees, when it is only necessary to take the chill off a house, a hot water system is especially well adapted to fulfill the requirements, and the temperature of the water may be kept as low as desired, whereas with the mercury seal vacuum system 170 degrees F. is about as low a temperature as can be maintained, and then not for any length of time, owing to imperceptible air leaks, which destroy the vacuum.

The advantage of quick heating in the vacuum system is, in a measure, offset by the advantage possessed by hot water for storing the heat during the night. With the mercury seal system all radiators are kept at the same temperature. The steam supply may not be throttled without fear of water backing up in the radiators and causing noise. In hot water heating systems the temperature of each radiator may be controlled at will by throttling down the supply, thus giving individual control of the temperature of each room.

SUGGESTIONS TO FITTERS FOR INSTALLING GRAVITY VACUUM SYSTEMS.

The same care must be exercised in draining and dripping the piping that would be necessary if it were intended to erect the apparatus without the use of the vacuum system. Extra precaution should be taken, however, to have all joints tight, and to have all fittings free from sand holes.

Care should be taken that all valves used are carefully packed so as to avoid the leakage of air into the system while the vacuum

is being maintained. The manufacturers' packing in the valves should not be depended on but should be removed and carefully replaced with lamp wick dipped in oil and plumbago, being sure that sufficient wick is used to make the valve tight around the stem. All check valves used should be swing checks, with asbestos or Jenkins' seats. Care should be taken to see that there is no leak of air into the system through the safety valve, water glass, gauge cocks or other trimmings.

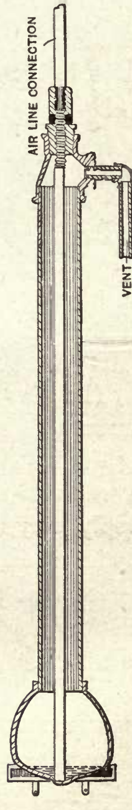

Fig. 44.—Mercury Seal.

The air pipe should not be less than ¼ inch between the air valve and the first fitting, where it should increase to ½ inch pipe. The first fitting below the air valve should therefore be a ¼ x ½ inch elbow in every case. No air riser should be less than ½ inch, and where the air riser extends above the second floor or is connected to more than two air valves should not be less than ¾ inch. The horizontal air main should be run on the basement ceiling, and need not be larger than 1 inch except in cases of extreme length, where it should be 1¼ inches.

In making up the air lines, it is recommended that galvanized fittings be used and that the joints on the air lines be made up of asphaltum and the fittings painted all over on the outside with asphaltum in order to close up any sandholes. Care must be taken to have the air piping tight. Care must also be taken to give the air piping a good pitch of not less than 1 inch to 10 feet toward the boiler. The air piping must contain no pockets or traps of any kind. The connections of the air piping to the air valves should be made with ground joint brass unions.

VACUUM AIR VALVE SYSTEMS.

Several patterns of vacuum air valves have been put on the market designed to permit the escape of air from radiators and to prevent its reentering them when the steam pressure falls. With a system perfectly tight at all valves, fittings, etc., and with

all vacuum air valves in working order a steam plant may be run as a vacuum system. It can derive the advantage of a great range in the temperature of the radiators by simply raising the steam pressure sufficiently to drive out all the air, for then when the pressure falls below that of the atmosphere the radiators will remain filled with steam at a minus pressure and at a temperature below the boiling point, viz., 212 degrees F.

Mr. George D. Hoffman writes of one of these systems as follows:

"The difficulty heretofore existing of being able positively and automatically to prevent the air from going back into the system when the steam pressure is reduced below that of atmosphere, is overcome by the use of the automatic air and vacuum valve and the air line system of vacuum heating.

"The valve is intended to be a vacuum system in itself. With an apparatus that is practically air-tight in all its joints and connections, simply screw on the valves in place of the ordinary air valves and you have installed a complete system of vacuum steam heating. The use of these valves does not necessitate air lines or any mechanical appliance for exhausting the air. Pressure exhausts the air from the system through the valve, and then when pressure goes off the valve automatically closes, preventing the ingress of air into the apparatus through the valve. The valve is especially designed for use in connection with residence work, stores and small apartments where the number of radiators in connection with any one plant is limited."

It is not wise to attempt to provide a vacuum system in large buildings by simply attaching vacuum air valves to the radiators, because of the great number of valve stuffing boxes, fittings, etc., at which an inleakage of air is liable to occur and destroy the vacuum, necessitating the raising of steam pressure at frequent intervals to force the air out of the system.

THE VAPOR SYSTEM OF HEATING.

The vapor system is a modified two-pipe system of steam heating, arranged with devices to prevent more than a few ounces pressure accumulating in the boiler or radiators. Each radiator is equipped with a special supply valve, designed

to admit a volume of vapor or low pressure steam sufficient to supply a portion or all of the radiating surface. At the return end of each radiator is placed a small combined water seal and air vent (see Fig. 46), designed to permit the escape of air and

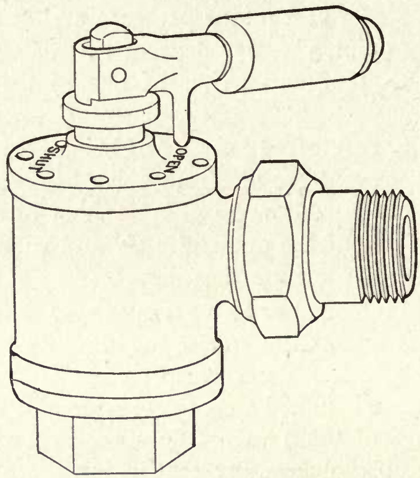

Fig. 45.—Vapor System Supply Valve.

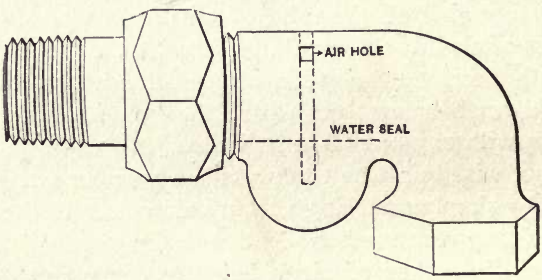

Fig. 46.—Union Elbow for Return End of Radiator.

the water condensed in the radiator. Fig. 47 shows a radiator equipped with the supply valve and the combined water seal and air vent. Radiators of the hot water type are invariably used in connection with this system.

The returns from the radiators, this being a two-pipe system, are combined in the basement and lead to a receiver (see Fig. 48) connected with the boiler. The main return is sealed at the end, as

illustrated in Fig. 49, to prevent the escape of vapor to the cellar. From the chamber C the air combined with some vapor from

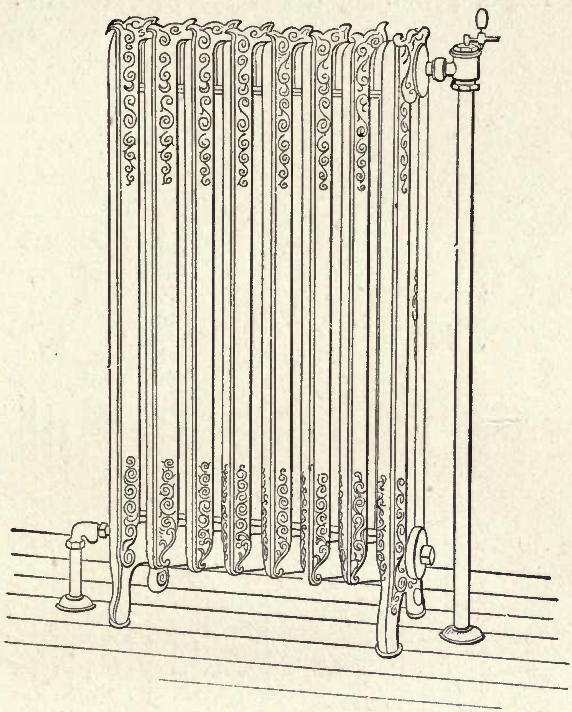

Fig. 47.—Radiator Connected on Vapor System.

the system escapes through the air line to the condensing radiators suspended from the basement ceiling as shown in Fig. 48. The vapor is condensed in these radiators and flows back by gravity to the boiler, the air escaping to the smoke flue. The latter is preferable to any other point of escape since the heat in the flue causes a slight pull on the air line accelerating the removal of the air from the system.

The receiver is open at the top, and there is no check valve between it and the boiler. It, therefore, acts as a perfect safety valve to prevent any excess of pressure in the boiler. Should the boiler pressure increase, the water would be driven out into the receiver. The float therein would be raised and the drafts

closed. Should the pressure continue to increase from any cause the float in the receiver would rise until the lever of the relief valve is raised, permitting the escape of steam and reducing

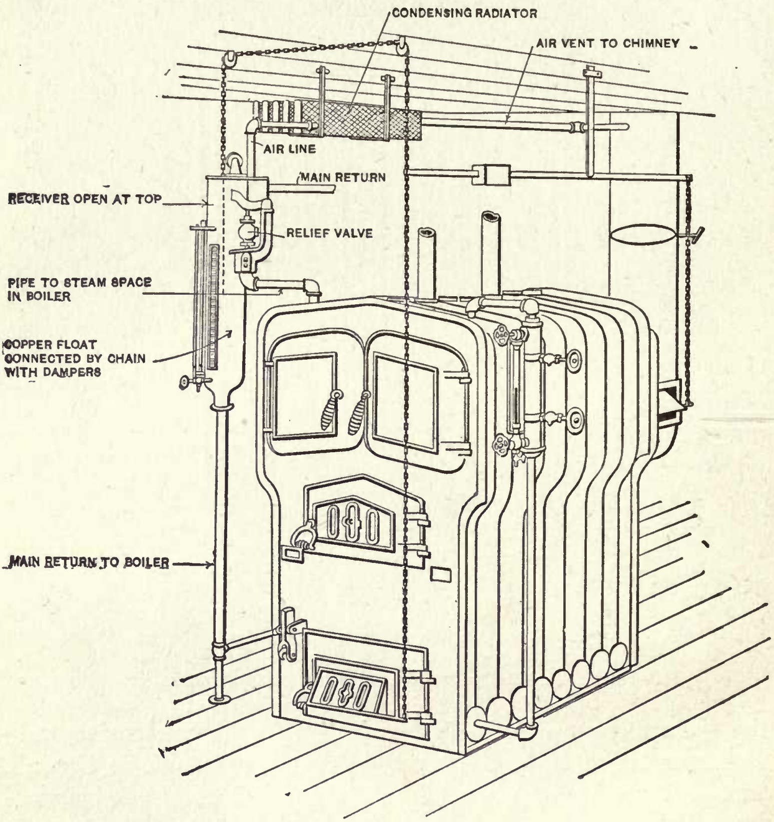

Fig. 48.—Connections at Boiler, Showing Condensing Coil.

the pressure. A glass water gauge is attached to the receiver, and a scale indicates the boiler pressure in ounces. No other pressure gauge is necessary.

The maximum pressure never exceeds 13 ounces, and therefore the size of the radiators must be based on relatively low temperatures, and an amount of surface within 10 or 15 per cent. of that

required with hot water heating is commonly provided to warm the rooms properly in the coldest weather. With this system one cannot overcome the effect of a shortage in radiating surface by increasing the pressure and temperature as in low pressure steam heating, since the water would be backed out of the boiler through the main return connected with the receiver.

The water line of the boiler should be at least 4 feet below the basement ceiling to give sufficient pitch to the pipes and to provide ample hight to cause the water to flow back into the boiler. A common arrangement of piping is shown in Fig. 50. The returns must be run overhead in the basement, that is, they must be "dry." These pipes are preferably left uncovered in order to promote the condensation of any vapor escaping to them from the radiators.

The vapor system may be used in connection with exhaust steam plants supplemented by live steam and with central heating plants, as shown in Fig. 51. When the condensation is not returned to the boilers the pump and receiver are omitted, and the condensation is discharged to the sewer through a cooling coil. Central station heating companies commonly require the cooling

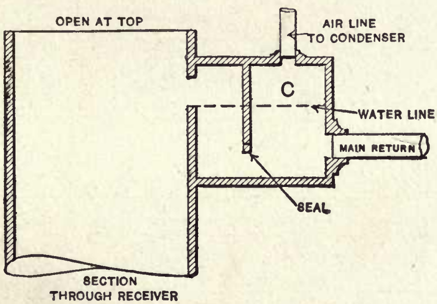

Fig. 49.—Seal at End of Main Return.

coil on low pressure systems to contain one-fifth to one-sixth of the entire direct radiating surface in the building.

In moderate weather the regulator is set to close the draft on 1 to 2 ounces, in severe weather 4 to 6 ounces.

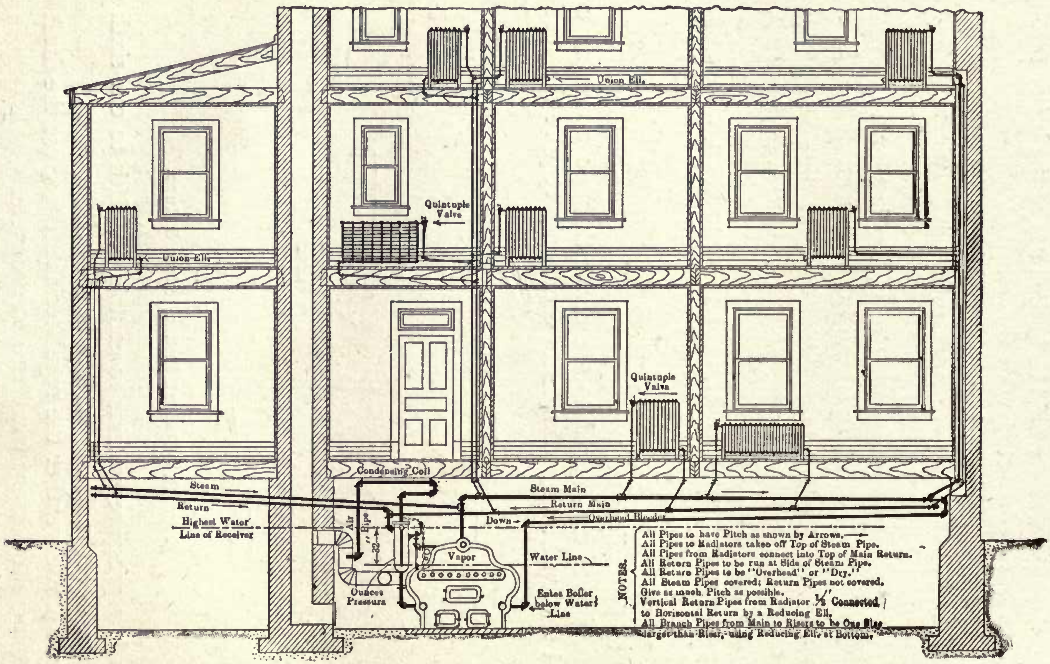

Fig. 50.—The Vapor System, Showing Manner of Running Steam and Return Pipes.

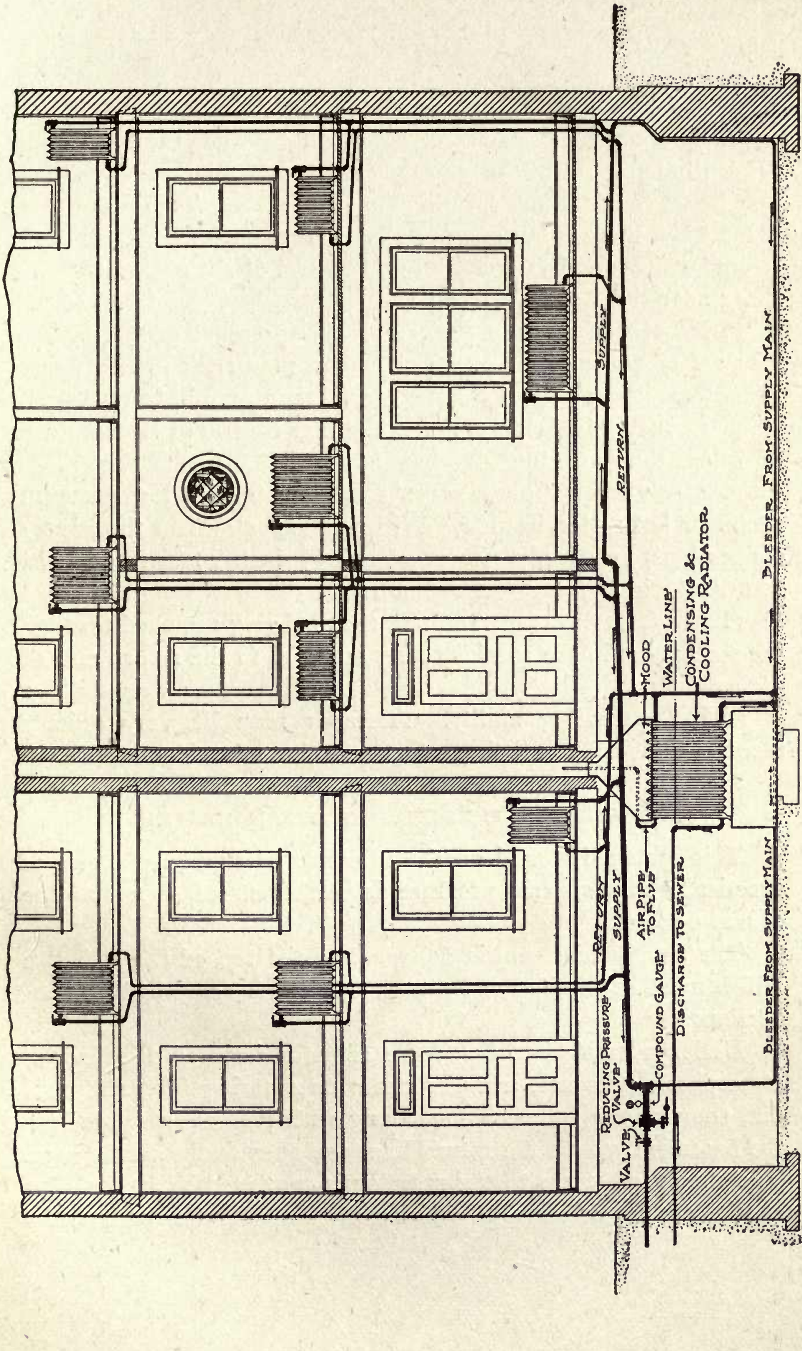

Fig. 51.—Broomell's Vapor System, Showing Piping with Supply from Street.

ADVANTAGES CLAIMED FOR THE VAPOR SYSTEM.

1. The control of the heat given off by each radiator independently by means of the quintuple valve shown in Fig. 45, which may be set to admit any desired amount of steam. This is of great value when there is but one radiator in a room, for with ordinary steam heating one has practically no control of the room temperature under these conditions.

2. Freedom from any danger of over-pressure on the boiler. The safety valve of an ordinary system may stick or the water in the expansion pipe of a hot water system may become frozen.

3. Economy in fuel because of the easy control of temperature afforded, thus avoiding overheating.

4. Much smaller pipes may be used than with low pressure steam or hot water heating. With the vapor system the supply connections practically never exceed ¾ inch in size, and the returns ½ inch for direct radiators.

5. Air valves are not required, the air being removed through the small vent in the special fitting attached to the return end of each radiator.

6. Quick heating ability. A vapor may be very quickly secured sufficient to fill the radiators without forcing the fire.

THE ATMOSPHERIC SYSTEM OF STEAM HEATING.

This system is based on the principle of supplying steam at practically atmospheric pressure in the form of vapor as the source of heat.

The supply of steam is delivered to the top of the radiator, which must be of the hot water type, with top and bottom connections.

Under normal conditions, the radiator is filled with air at atmospheric pressure, and the steam on entering forces this air out, through the return piping along with the condensation into the basement, where the return lines are open to the atmosphere. The supply of steam to the radiator is regulated by means of a specially constructed valve, which will admit the desired quantity

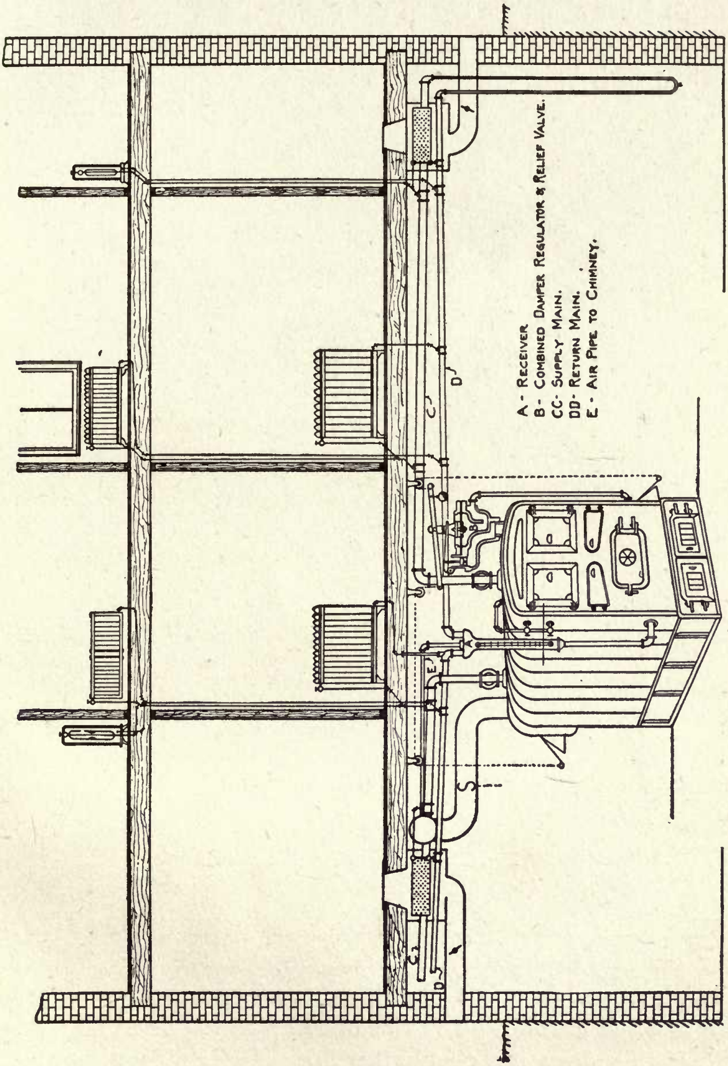

Fig. 52.—Interior Steam Piping. Atmospheric System. Independent Boiler.

of steam to the radiator. When the steam is admitted, it spreads in a thin layer along the top of the radiator, and as it changes to water, runs down the walls of the radiator, heating them and becoming cooled.

This system as applied to individual plants is illustrated in Fig. 52.

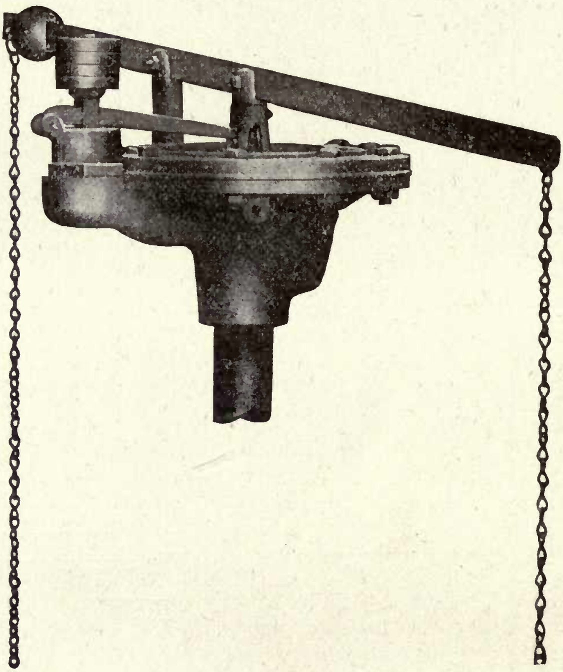

Fig. 53.—Damper Regulator.

When the boiler pressure rises above that desired (five to eight ounces) the water is forced from the boiler into the stand-pipe, which raises the float, causing the draft door to close and the check draft in the smoke pipe to open. (See Fig. 54.) If this does not check the fire sufficiently the float will continue to rise, causing check valve at H to close, preventing standpipe overflowing. In addition to the above is the safety valve on the boiler.

No valve is used at the return end of radiators.

No air valves are required.

Since a portion of the radiator acts as hot water radiation, about 25 per cent. more radiation must be used than would be required with steam radiation.

In this system "the control of steam supply to the radiator is accomplished by means of a specially constructed valve, and is the only appliance which is made particularly for the system.

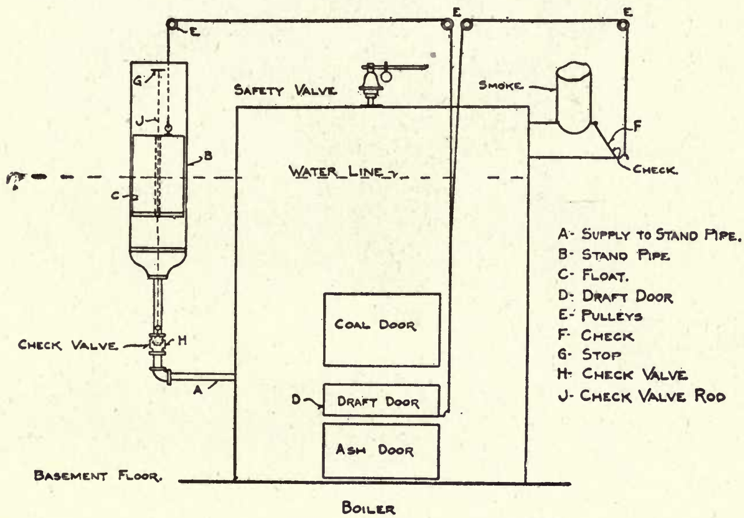

Fig. 54.—Damper Regulator Atmospheric System.

"This valve is made only in ¾-inch size. The opening in the valve seat is proportional to the size of the radiator which the valve is to serve, and the amount of surface in the radiator must always be specified when ordering valves. For instance, a valve constructed for a 40-foot radiator cannot be used properly on any other size, and if put on an 80-foot radiator, would only admit steam enough in maximum open position to heat half the radiator.

"Steam is admitted to the radiator by opening the valve, turning to the left in the usual manner.

"It is desirable at periods of maximum demand to heat the

upper four-fifths of the radiator; this will have to be determined by experiment, as no two radiators of the same size are set under the same conditions. The valve opens full in four turns, one complete turn will open it one-quarter, two turns one-half, and so on."

CHAPTER X.

HOT WATER HEATING BY FORCED CIRCULATION.

In this chapter extracts from several articles relating to large installations are given which bring out a number of interesting features.

One of the arugments frequently advanced in favor of hot water heating in connection with condensing plants is that the condensing water from the surface condenser may be circulated through the heating coils and is sufficiently hot to keep up the temperature in the buildings in mild weather. Of course the outside temperature at which it is necessary to change from condenser water to the heating of same by live steam depends on the proportioning of the radiation. During cold weather the surface condenser is operated entirely with cold water and the water for the heating system is passed through a live steam heater.

A velocity of flow of 400 feet per minute is about the maximum commonly used.

CENTRAL HOT WATER HEATING PLANTS.*

"The equipment and operation of the system is, in brief, as follows: The steam boilers, engines, and dynamos are such as may be used in the ordinary electric light station. Heaters of the tubular type, through which the water passes from the pumps to the mains, receive the exhaust steam from the engines, heating the water to any desired temperature. When more exhaust is being produced than is required to heat the water, the excess is delivered to a water-storage tank to be used later when the electrical output is small. The circulating system consists of two wrought-iron pipes, laid side by side in the ground, carefully protected by insulation (See Fig. 55), one pipe for the outflow of hot water

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* Extracts from paper by H. T. Yaryan, A. S. M. E., Cincinnati meeting, 1900.

impelled by the pumps, the other for the return water from the coils in the various houses heated, going back to the suction end of the pumps, to be forced again through the heaters, where the loss in temperature is restored. The heaters used are surface condensers.

The houses are equipped with radiation sufficient to heat them to a temperature of 70 degrees F. when the outside temperature is freezing, with water entering at 160 degrees. By raising or lowering the temperature of the water one degree for each degree of variation in the outside temperature, it is stated that a constant temperature may be maintained in the houses in all kinds of

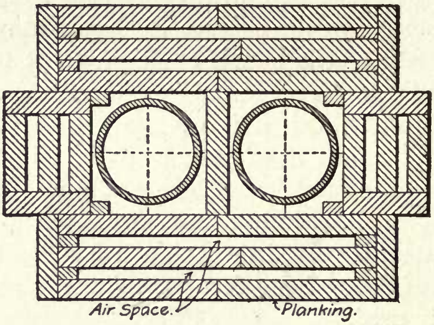

Fig. 55.—Arrangement of Insulation.

weather. The extreme limits of the water temperature is 130 degrees in moderate weather and 212 degrees in the coldest. The water reaches the extreme end of the lines, three-quarters of a mile from the station, in the coldest weather, with a loss of 12 degrees F., which would be an average of 6 degrees to all of the houses. As the water returns to the station with a drop of 35 degrees, this would indicate about 17 per cent. loss in the ground, which the author thought was about correct. A pressure of 60 pounds is maintained on the feed line during cold weather and 40 pounds during moderate weather. The service pipes to the various houses are 1-inch pipe, and the return line is throttled with a disk inside the building, the size of opening depending on the quantity of radiation, but averaging ⅝ inch."

As to the corrosion of mains, the author had this to say:

"10,000 feet of pipe were relaid last summer, after being in use five years. The wrought-iron pipe showed no signs whatever of corrosion, but steel pipe was perceptibly affected. The pipe showed the formation of nodules on the inside, varying in number from a dozen to 100 in a length of pipe, and in size from a grain of corn to a hazel nut. When the nodules were removed, a soft under layer of carbon remained, and with the finger nail this could be removed, developing a perfectly round pit from 1/8 to 3/16 inch deep; but in no instance had the pit perforated the pipe. As a matter of curiosity, some steel pipe which had been in use two years was examined, and found to be in exactly the same condition as that of five years' use. The action seems to be electrolytic, some particle of impurity forming the nucleus for decomposition of the steel. If any inference can be drawn from the facts stated, it is that the impurity has lost its power to cause electrolysis after being surrounded by a sufficient amount of graphite."

HOT WATER HEATING BY FORCED CIRCULATION.*

"A type of heating system that is gaining in favor as its advantages are becoming better known is hot water heating by forced circulation. By this is meant a system of hot water heating in which a circulation is induced by means of a pump placed in the circuit of the mains, the water being heated by either exhaust or live steam, or both. The advantages claimed for this system are economy in steam consumption, ease of control, the maintenance of a constant temperature in the building irrespective of the outside temperature, and the ability to run the mains anywhere regardless of grades, thereby permitting the location of the power house at a desirable location. This is often impossible with a steam system unless pumps or other devices are installed to return the condensation if it is desired to save it, or unless expensive tunnels are built.

The disadvantages of this system are that it is necessary to have an independent steam system if steam is required for other

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* Extract from an article on "The Choice of Heating Equipment for Manufacturing Plants," by G. W. Stanton, in *The Engineering Magazine.*

purposes (but as before stated, in my mind this is a wise provision and one that conduces to economy in operation); that it requires a greater amount of heating surface than a steam system, and a consequent greater first cost; that it requires greater engineering ability or knowledge in design (this fact may be disputed, but I know from personal experience that a great deal of the prejudice against this system has been caused by the knowledge of the unsuccessful operation of poorly designed hot water systems); and that it demands more careful and better installation than the usual type of steam system, to prevent leaks and consequent damage.

It also uses power for the operation of the circulating pumps, but in this it is on a par with vacuum systems and (except in small installations) smaller in its demands than hot-blast systems.

Owing to the circulation being forced, the pipe sizes can be very much smaller than in any type of steam system; where the mains are long this is a distinct advantage. The radiation required is usually 12 per cent. in excess of that required for vacuum steam systems, and 20 per cent. above that of gravity low-pressure steam systems.

One of the great advantages of this system—and one that deserves consideration—is the ability to control the temperature in the buildings at the power house, besides the individual control of the radiators or coils. That this means much will be conceded when one considers the great variation in temperatures to which residents in the climate of eastern North America are subjected to. In New York City, for example, the temperature during the heating season ranges from zero to 50 degrees and even higher. As sufficient radiation must be installed to heat the rooms to 70 degrees in zero weather, and as the average temperature for the heating season in New York City is approximately 35 degrees, therefore with a steam system 100 per cent. more capacity must be provided for heating than is normally required. Of course, if proper attention could be given to a steam system and the radiation were shut off when the temperature gets above 70 degrees, this excess use would not occur; but unfortunately this close attention is rarely given where there are a great number of radiators or coils, and besides it is so much easier to open a

window and let the surplus heat escape, and this is what is usually done.

In the better class of buildings, thermostatic valves are installed on the heat sources, and these regulate the temperature of the rooms by regulating the supply of steam to the radiators; but as such devices are an expense to install and expensive and troublesome to maintain, their installation has been confined generally to the better classes of schools, office buildings, hotels and public buildings, and where first cost is not the only consideration.

It may be asserted that it is possible to obtain a degree of regulation with steam by varying the pressure; but the best results obtainable by such a method depend on the use of high pressure steam direct from the boiler, and even then it is not advisable to use a greater pressure than 30 pounds in cast-iron radiators. This would give an increase in the temperature of the steam from 212 degrees to 274 degrees, and if exhaust steam is used, would involve the increase of the back pressure on the engines, which is certainly inadvisable unless the engine be small or the engine load light.

The hot water system is especially applicable to the heating of many widely separated buildings supplied from a central power plant, and where ventilation is required the fans are driven by motors. If hot water is required for lavatory, domestic, or other purposes, that would have to be furnished by a central hot water lavatory system, the water for this system and also for the heating system being heated by the exhaust steam from the power units and all apparatus centralized in the power house.

This mode of heating has been installed to operate in connection with condensing engines and turbines, the vacuum on the engines or turbines being reduced or increased to correspond to the outdoor temperatures. For instance:

Temperature outdoors. Degrees.	Vacuum in inches at engine.
0	0
10	10
20	16
30	20
40	24
50	26

From the reports of the U. S. Weather Bureau* the following table was compiled, giving the number of hours of different degrees of temperature during a ten-hour working day and on Sundays and holidays during an entire heating season of 200 days.

TABLE XLII.

Outdoor temperature, degrees.	Hours of heating, working days.	Hours of heating, nights, Sundays and holidays.	Total hours per season.
0-10	5	63	68
10-20	95	220	315
20-30	320	675	995
30-40	465	1,124	1,589
40-50	400	920	1,320
50-60	260	270	530
	1,545	3,272	4,817

The time of heating per working day is based on ten hours per day during which exhaust steam is available; during the remainder of the time live steam would have to be used.

Where it is possible to operate the engines condensing, such a combination is certainly most attractive and offers great possibilities in saving of coal through the reduction of steam consumption of the engines during the day, in the saving of live steam, when exhaust steam is not available, by using only just enough live steam to maintain the actual temperature required."

EVANS-ALMIRALL HOT WATER HEATING SYSTEM.†

"In this system, water is set in motion through a series of pipes by means of a centrifugal or other pump and from the pump is carried through an exhaust steam heater and if desired through an economizer, and possibly in addition through a live steam heater. Any combination of heating devices may be used as the circumstances of any case require.

From the heating devices the water is circulated through mains to the various radiators and from these is brought back in a single return to the pump.

Any form of closed feed-water heater or condenser may be

* These temperatures and hours are taken from the report of the U. S. Weather Bureau at Scranton, Pa., and are representative of a large section of the United States.

† Extracts from article in *The Engineer*, January 1, 1906.

used, the steam being sent through the tubes and the water circulated around the outside of the tubes in the usual construction. The heater may be inserted in the exhaust line of either a condensing or non-condensing plant and the condensed steam may be returned to the boiler or allowed to go to waste as is most economical.

By the series arrangement of exhaust heater, economizer and live steam heater, it is possible to heat the water to any desired degree of temperature, and by regulating the amount of live steam used, there is no waste, as just sufficient live steam can be used to produce the desired temperature. Exhaust and live steam are never mixed, as the two heaters are entirely separate, so that live steam can always be returned to the boiler without the use of any purifying apparatus.

In the construction of the pipe system, since the flow is forced by means of a circulating pump, the grade of the overflow and return mains is a matter of no importance and can be made to suit the convenience of the system.

Double- or single-pipe systems may be used, according to circumstances. In the double-pipe system, mains and returns are carried to each radiator and circulation is effected from one system to the other. In the single system, the water is passed through one pipe which forms a loop from the pump to the district to be heated and back again to the pump, connections being made to various buildings or radiators by shunts to and from the single pipe.

In underground construction, full-weight wrought-iron pipe is used with screwed cast-iron fittings for service connections. These fittings are provided with valves and enclosed in brick boxes, so that they can be easily gotten at and new connections can be made without disturbing the mains. Slip expansion joints are provided to take care of the go and come in the pipe, and these are also encased in brick boxes, anchors being provided by means of wrought-iron bands and masonry so that the pipe is forced to take up its expansion at the joints provided.

For covering the underground mains, Wykoff wood stave is used.

The radiators are of the regular hot water type connected on

the single-pipe system. The connections are from opposite ends of the radiator with a down-feed riser, the radiator valve being placed in the return connection.

In the system described it is stated that the entire system has about 30,000 square feet of radiation and requires 12 horse-power at the circulating pump. The temperature of the circulating water is varied to suit conditions between 100 and 185 degrees and at the same time no change in the back pressure on the engine is ever produced. By the utilization of heat from the flue gases, a considerable economy is effected. The piping is arranged so that each particle of water travels the same distance from the time it leaves the pipe till it returns, thus avoiding short-circuiting."

"The water is heated by exhaust steam from the engines,* which passes through tubular heaters of a construction similar to that used in the ordinary closed feed-water heater. They are so connected to the exhaust piping that steam may be passed through them singly or in series for moderate heating, or in multiple for colder weather. Their construction and piping arrangements are such that they may also be used for heating when the engines are operated condensing, in which case the exhaust passes through them before entering the condenser. The vacuum is then varied in the condensing system to suit the outside temperature, the circulating water in the heating system being maintained at a temperature within ten degrees of that of the steam at the existing vacuum.

The hot water is passed through the heating coils and radiators in the buildings by the turbine-driven centrifugal pumps. One of these units is operated continuously when heating is required. The pumps are operated at a comparatively high pressure for this service, inasmuch as the volume of flow is large and the distances covered are comparatively great. The systems of heating coils in the various buildings are connected in multiple or parallel.

The hot water may be passed through either heater singly, through both in series, or through both in multiple. An auxiliary live steam heater is provided for use in heating the circulation to higher temperatures in case the exhaust steam is not sufficient

* Extract from an article in *The Engineering Record*, April 22, 1905.

in quantity or is unavailable; this heater may be by-passed or cut into the circulating line by handling three valves. An air trap is set at the point where the line leaves the power house for freeing the system of entrained air."

VOLUME OF WATER REQUIRED FOR HOT WATER HEATING SERVICE.*

"Each square foot of hot water radiation in the city will require approximately one gallon of water per hour. It is very certain that some plants are designed to supply less than this amount, but in such cases it requires a higher temperature of the circulating water and allows little chance for future expansion of the plant. A drop of 20 degrees, i.e., 20 B.T.U. heat loss per pound of water passing through the radiator, is probably a maximum and indicates the minimum amount of water that should be circulated. In practice, this heat loss would probably be nearer 15 B.T.U. per pound, and consequently, would necessitate the use of somewhat more than one gallon of water per square foot of heating surface per hour. All things considered, the above italicised statement will satisfy every condition. Having the total number of square feet of radiation in the district, the total amount of water circulated through the mains per hour can be obtained, after which the size of the pumps in the power plant may be estimated. . . .

The district is first chosen and the layout of the conduit system is made. This is done independently of the sizes of the pipes. When this layout is finally completed, the pipe sizes are roughly calculated for all the important points in the system. . . .

In some cases, when close estimating is not required, it is satisfactory to assume a velocity of the water and find the diameter without considering the friction loss. In many cases, however, this would soon prove a positive loss to the company. With a low velocity, the pipe would be large, the first cost would be large, and the operating cost would be low. On the other hand, if the velocity were high, the pipe would be small, the first cost would be small, and the operating cost and depreciation would be large. . . .

*** Extract from "Handbook for Heating and Ventilating Engineers," by Prof. J. D. Hoffman.**

It is found in plants that are in first-class operation that the velocities range from 5 to 7 feet per second."

HOT WATER HEATING BY FORCED CIRCULATION.*

Hot Water vs. Steam for Mill Heating.

"Because of the exceptionally high specific heat of water, as well as its relatively high specific gravity, it possesses special value as a medium for the storage, as well as for the carriage of heat. Also, because its rate of flow through heaters, either radiators or circulation coils, can be closely regulated by a single valve, a water system furnishes better means than a steam system for temperature regulation. A system having 10,000 square feet of radiating surface will contain from seven to eight tons of water, and a change of 30 degrees in the temperature of that water involves the transfer of nearly 500,000 thermal units, or an hour's loss of heat through 36,000 square feet of 18-inch brick wall in zero weather. Water is thus to a heating system what a large fly-wheel is to an engine. It stores up the surplus when a surplus exists, and it carries over and yields that surplus when the supply at the source drops or fails. By adding a storage tank of 250 cubic feet capacity for each 10,000 square feet of heating surface used, the storage value of the water in the system may be doubled.

The cost of installing a hot water system must exceed that of an equivalent steam system, for the reason that radiating surfaces must be larger and because power must be used for mechanically circulating water. The power necessitates engines and pumps and some small outlay for steam, the steam, however, not being lost, as it may be turned to account in heating the water it circulates. The engine and pump work also ultimately resolves itself into heat, so that nothing is lost. The mechanical circulation of water makes a rapid flow and use of small pipes practicable in mains and branches. It eliminates the necessity of providing and carefully maintaining such carefully made grades in pipe runs as are essential to water flow in gravity work. Water may be sent

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* Extracts from paper on "Warming and Ventilating of Mills," by Prof. S. H. Woodbridge, N. E. Cotton Mfg. Association, April 25, 1900.

in any and every course under the action of a centrifugal or screw pump. In a paper prepared for the purpose of promoting economy in methods and practice the plunger or piston direct acting pump can be mentioned only to hold it up to condemnation.

The advantages of water over steam for heating work are in a measure offset by the fact that power is essential to the circulation of water, whereas steam, even at low pressure, is automobile. For night or shut-down heating work beyond the capacity of the system's storage, a use of water necessitates the use of power, though in small amount, approximating a range of from 500 to 1,500 foot-pounds per cubic foot of water circulated.

In the summing up for water as against steam it may be said: First, that by its use the engine exhaust may be condensed and the water heated without back pressure effects, the heater playing to some extent, at times, the part of a condenser. Second, that when required, as in extreme weather, the water so heated may be further heated by live steam surfaces, either in the same or in another steam heater or by a furnace heater. Third, that the water within pipes and the storage tanks conserves heat by storing it. Fourth, that the temperature of the entire body of water is made perfectly controllable, either by hand or by automatic means. Fifth, that room temperatures are made easily controllable by regulating, by means of a single valve, the water flow through the heaters within any room or apartment. Sixth, because of its heat storage capacity, water can maintain mill temperatures after shut-down as steam can not. If equal volumes of water and of steam at 212 degrees be cooled to 32 degrees, the water yields 257 times the heat yielded by steam. Seventh, that the heating action of water is more steady, or less fluctuating than that of steam and that by as much as the temperature of the water surface is lower than that of steam, the quality of water heat is better than that of steam. Water heating by forced circulation is cheaper than steam only as waste may be conserved and made usable. In the initial cost and maintenance of the plant the water is slightly the more expensive of the two. A water plant is wholly unsuited to large mill work in which heating is required when power is not available."

HOT WATER HEATING IN THE SOUTH STATION, BOSTON.*

"Hot water was chosen for the circulating medium only after a thorough study of the possibilities and limitations in the use of both steam and hot water. This made it evident that a hot water circulating system in the form of a loop with return mains in transverse subway not only overcame all physical difficulties encountered in the problem of distribution, but under the rather unusual conditions which surrounded the heating plant, was superior in every respect. The rapid circulation secured by the use of pumps made all points in the system equally accessible and easy to heat. The amount of water necessary to supply the heat required could be conveyed in a flow main no larger than 8 inches in diameter, and without difficulty due to unavoidable pockets at certain points, notably at the main suburban exit.

Owing to the storage value of the large body of water contained in the system, the surplus of waste steam at periods of heavy load could be absorbed, within limits, for use when there might be a deficiency of exhaust at times to reduction of load, and be utilized to a greater extent than possible by other methods. Furthermore, while all heating apparatus must be adequate for the severest weather conditions, the hot water system possessed the considerable advantage of being capable of operating at lowered temperatures corresponding to mild or moderate weather, thus preventing waste and discomfort from overheating and undue radiation and leakage losses.

The water is circulated by one of two 8-inch centrifugal pumps, each driven at a speed of about 375 revolutions per minute by an 8½ by 8 Westinghouse standard engine. The water reaches the pump suctions through the 8-inch return main by way of transverse subway, and is delivered to the supply side of the system after becoming heated to a suitable temperature in a central heating plant located near the circulating pumps in the power house. This consists of three specially designed heaters, two being for exhaust and a third and smaller one for the use of live steam whenever there may be an insufficient supply of

* Extract from paper by Walter C. Kerr, A. S. M. E., Dec., 1899.

exhaust. Two exhaust heaters were used largely because of the inexpediency of installing a single heater of large dimensions, and partly on account of the advantage of having the principal part of the heating plant in duplicate. There was also advantage in regulation, by subdivision. The exhaust steam heaters are capable of condensing 24,000 pounds, and the live-steam heater approximately 18,000 pounds steam per hour, live steam being admitted to the heater only to supply deficiency, the quantity being regulated by an automatic thermostat valve.

The heated water passes from the central heating plant into the flow main and to the exterior circuit, which forms a loop over five-eighths of a mile in length. This entire system, in which there is necessarily a large amount of expansion and contraction, contains no expansion joints, the design of the piping and the free use of long radius pipe bends permitting the strains due to expansion to be taken up without imposing undue stress on fiber or joint. All turns in the flow and return mains were made by these bends, no elbows being used, thus materially reducing the circulating head, which would otherwise have been excessive except where larger and more expensive mains were provided. The engineering problem involved in the proper design, support, and anchoring of these hot water mains was not the least interesting of the several somewhat unusual features of the work. In this connection, it is interesting to note that the actual circulating head proved to be within 2 per cent. of the estimated amount."

HOT WATER HEATING FOR MILLS.*

"In order to obtain satisfactory results with the hot water or forced circulating system, the speed of the water through the circulating pipes must be high to prevent any too sluggish movement in the shunts or isolated coils that offer a little more frictional resistance than the direct lines. The temperature should also not be allowed to fall very much during its passage through the entire system to secure uniform efficiency throughout. In ordinarily cool weather it should be sent out as warm as 200 degrees F., and returned at not less than 150 degrees F.

" Extract from a paper by A. G. Hosmer, presented before the National Association of Cotton Manufacturers, in 1908.

In some plants this system has been installed with a view of utilizing the heat of the exhaust steam from condensing engines on its way to the condenser, by placing a heater between and running the engines with a vacuum more or less reduced. If, as has been mentioned, the temperature of the outgoing water should be about 200 degrees F. and that entering the heater 150 degrees F., it is evident that to enable it to absorb any heat from the exhaust, the vacuum would have to be adjusted to make the temperature of said exhaust somewhat above 150 degrees, or at least 160 degrees F.; this corresponds to a vacuum of about 20 inches, or 5 pounds, absolute pressure. At a temperature of 180 degrees F. we have only 12 inches, and at 200 degrees F., 6 inches, or a little less than one-fourth of the usual efficiency of the condenser. The choice is offered then to run either a low vacuum and sacrifice a portion of the benefit derived from the condenser, or to add live steam in a secondary heater to make up for what the low-pressure steam cannot do.

For efficient results with this method of heating the water while passing through the exhaust chamber, the size of engines and capacity or requirements of the circulating system should be so proportioned that the passing water will absorb the entire amount of available heat in the exhaust steam. To illustrate, suppose the engines are running with a vacuum of 12 inches to give the exhaust a temperature of about 180 degrees F.; our circulating water enters the heater at 150 degrees F., thus we have 30 degrees F. available. If the quantity of water exposed in the heater is sufficient to absorb all the heat passing in the exhaust (that is, heat above 150 degrees F.), all is well, but as the temperature of this heating medium is governed wholly by the condition of the vacuum, regardless of the quantity of exhaust going by, any excess will pass through without giving up its available heat, and the impaired vacuum will have to be made up by steam from the boilers, for a part of which there will be no return. In other words, enough extra heat in form of boiler steam will have to be added to raise the mean effective pressure to the point it would be with regular conditions in condenser, and as a return only what heat has been absorbed by the circulating water will be received.

Every 2.04 inches taken from the vacuum decreases the mean effective pressure 1 pound. It will be remembered that the back pressure or vacuum, as the condition on the discharge side of an engine may be called, applies to the entire length of stroke the same as mean effective pressure. It is therefore evident that the raising or lowering of this pressure, even a few pounds, on a large engine cylinder means quite a serious difference in the important work done by the condenser and the power output of the engine.

If the quantity of exhaust steam is insufficient at times, care must be taken to avoid lowering the vacuum too much, as it will spoil regulation, and complicate things generally, especially with compound engines. The line between waste and economy in this proposition is so finely drawn that to use the minimum temperature of the exhaust with also the minimum amount of direct steam and realize the best conditions, the adjustments should be wached very closely.

In installations where the engines and condenser systems are not considered and direct steam used entirely, the heaters are usually placed in the boiler house enough above the water line of boilers to insure a gravity return for the condensation. Provision must, of course, be made for operating the circulating pump at night and during holidays and Sundays. In nearly all equipments of this kind a supplementary heater or section of one of the heaters is used for the exhaust from the pump engine or turbine.

The most severe duty on the system comes when the plant is shut down, particularly early mornings. At such times the water must be used at the maximum temperature, and as the exhaust from engine is not over 210 degrees F., unless back pressure is made use of, the steam will pass by the water coils, transferring little or no heat to them. In places where condensing water is available it is sometimes better to install a small condenser for the pump engine and to do away with the inefficient exhaust heater.

The circulating system is sometimes connected with the economizers to use the waste heat passing to chimney during nonrunning hours. While the theory of this is excellent, in practice

it is hardly ever desirable, for the reason that the economizers during working hours are handling water quite hot and at a high pressure, and if cooler water at a much lower pressure is introduced it contracts the joints and produces a change in conditions which usually develops more or less serious leaks and perhaps the loss of a gasket when the high pressure is again admitted. The repairing of economizer joints is not a pleasant task, to say nothing of the trouble and expense of having the machine out of commission when needed.

This system cannot be hurried to any extent, and it is consequently advisable to avoid letting a building cool down very much during cold weather by running the circulation continually. In this respect a direct-heating system using steam has a material advantage, as it is seldom needed until a few hours before starting time and rarely through the entire day.

In case of accidents to the machinery driving the pump a bad feature presents itself, as it is necessary to use steam in the circulating pipes instead of water. This is almost sure to cause trouble if the pockets or drops in the lines are not properly drained, as the entrapped condensation will cause hammering with innumerable leaks and frequently split some large fitting. Particular attention should be always given to installations of this kind to arrange the pipes with the same pitch and draining facilities that would be required if high-pressure steam was used."

STEAM VS. HOT WATER FOR CENTRAL HEATING SYSTEMS.

(The *Author.*)

While a number of features incident to the system of hot water heating by forced circulation favor its adoption in certain cases, there are features in connection with steam heating systems, especially the vacuum system, which commend themselves and which are peculiarly valuable in the overhauling of old plants and the substitution of a better system. As a rule old buildings are provided with steam radiators, viz., those having the loops nipple-connected at the bottom only. Radiators of this type are not adapted to hot water heating. On the other hand hot water type

radiators, viz., those with loops connected both top and bottom, are well adapted to steam heating.

With this system the steam, either live or exhaust, is delivered directly to the radiators, there condensed and the water of condensation discharged directly to the sewer through cooling coils in the basement or returned to the power house to be again evaporated in the boilers. Where no back-pressure is permissible or where the runs are long and trenching must be shallow, the two-pipe vacuum system may be used. With this system the condensation may be lifted if necessary.

Practically no pressure is carried in the radiators, whereas with the hot water system a heavy pressure is not uncommon.

Repairs are easily made to a steam heating system.

Although the supply lines may be made smaller in forced hot water systems than in steam systems, the returns in connection with the latter, especially with vacuum systems, may be made very much smaller than the returns from a hot water system. The mains, both supply and return, must be of the same size where they leave and enter the power house in hot water systems.

A single hand control valve is the only one necessary on each radiator with either the hot water or vacuum steam systems.

The automatic return valve in the latter takes the place of the air valve commonly used on radiators heated by hot water; the automatic return valve mentioned serves both as an air valve and a condensation remover.

It is in the power house that the simplicity of the steam system is apparent, as here the pressure reducing valve takes the place of the exhaust and live steam heaters used with the hot water system, and a steam-driven vacuum pump, with relatively small pipe connections, takes the place of the centrifugal pump driven by an engine or motor.

A quicker change in heating effect can be secured with steam than with hot water. Steam may be shut off for hours at a time during sunny days when the outside temperature runs up and in mills and plants not operated Sundays the heat may be shut off a greater number of hours than in the case of hot water owing to the longer time required to reheat the building after a shutting off of heat than in the case of steam.

With the vacuum system a partial heating of radiators may be secured in mild weather, each occupant having control of his room temperature. With hot water the temperature at which the water is delivered from the heaters in the power house is regulated by hand according to the weather, based on a chart giving water temperatures necessary to meet given weather conditions.

By this method it is necessary to send the water out at a temperature high enough to heat the most remote points and the most exposed rooms. Since the water loses temperature constantly from the time it leaves the power house, in the case of very long runs the buildings near the power house are likely to be overheated unless this effect is compensated for by allotting radiators more liberally at remote points.

With vacuum steam heating the radiator temperatures are the same in all buildings regardless of the distance from the power house.

In the case of old buildings heated by independent hot water heaters, if for any reason it is desired to continue to use water as a heating medium, tubular heaters may be placed in the basement and supplied by steam, thus heating the water.

As to using eduction water from condensers for hot water heating, since the temperature of this water is, say, 100 to 110 degrees or thereabouts, and must come back to the power house 20 degrees or so cooler, in order that the system shall be at all efficient it is evident that the low temperature of the radiators under these conditions limits the use of this method to mild weather conditions, unless the radiating surface be made inordinately large in proportion to the space heated. As the weather becomes colder heating by condenser water must be abandoned and the water be heated by live steam, assuming that the plant continues to run condensing, otherwise, if run non-condensing, the exhaust steam may be utilized.

In hot water systems where the heating is not continuous, there is a considerable loss due to the cooling of the large volume of water in pipes, radiators and heaters when heat is no longer wanted.

Danger of damage from freezing is of course much greater in hot water systems than with steam. The first cost of a steam heating system is as a rule considerably less than for hot water.

CHAPTER XI.

CENTRAL STEAM HEATING PLANTS AND MILL HEATING.

.. This chapter is devoted to the gist of a number of reports and articles on central steam heating plants. The reader is referred to the last article in the preceding chapter, which article relates to this chapter as well.

CENTRAL STEAM HEATING PLANTS.*

What to do with his exhaust steam is a question for the manager to decide. Which is best: to install condensers and thereby increase the efficiency of the engines 20 per cent. or more, and the capacity as well, or to sell this by-product as it is, in the form of heat? If the rates obtainable are fair, that is, if the heat can be sold nearly on a live steam basis, the plant should receive from the latter course at least three-quarters of the original value of the steam; while a condensing outfit would save but one-quarter to one-third of its value. If water for condensing is not available the argument for district heating is strengthened. Very many plants have short and sharp winter peak loads; and it would seem that under such conditions, their managers might be warranted in installing some inexpensive, simple, non-condensing engines, the exhaust from which would be well utilized for heating purposes, or it might even prove to be best, under proper conditions of load and fuel costs, to install condensing apparatus which would be run during the summer, but which in the heating season would be cut out, the exhaust steam then being utilized on the heating system. The use of the exhaust steam for heating will necessitate the carrying of some back pressure on the engines, thereby reducing both capacity and efficiency, a point not to be lost sight of.

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* Extracts from Report of Committee on District Heating, National Electric Light Association, Denver, 1905.

Before making the investment, the manager desires to know what income he may expect and how many square feet of radiation his exhaust steam will care for. Knowing the pounds of water per horse-power per hour consumed by his engines, and referring to his load curve of the previous winter, he can readily figure the minimum amount of exhaust steam that can be counted on for any single hour, and this should be the basis of his figures. Now come certain deductions: feed-water heaters and station leakage will probably take 15 per cent., while losses in mains and services may be as much more, though on the latter point we can cite one plant where the losses are less than 3 per cent.

The returns that we get from steam heating plants indicate that it is safe to figure for ordinary conditions about .20 pound of water per hour per square foot of radiation (varying from .05 to .50). As all the consumers are not using the steam at the same time, particularly if on a meter basis, the load factor should be no more neglected than in the lighting business. Naturally, with the heating business it is higher; and if we use the figure of 85 per cent. we shall probably be on the safe side.

Together with the above must be figured the interest charges on the total underground installation and that portion of the central station plant which is used for heating purposes; and lastly, depreciation—an item that is often neglected but which should nevertheless be estimated. From the best evidence at hand the last item seems to be about 5 per cent. per annum on the total cost of the investment.

It is a well-recognized fact that a district heating system has often enabled a company to secure contracts for light and power which it would not otherwise have obtained. These then produce an increase in load, which in turn supplies more exhaust steam to sell for heat.

Though it is generally considered good practice to limit the heating business to that which can be taken care of by the exhaust steam, in many cases live steam has to be supplied at certain times in order to give good service. Where this condition exists and the demand on the live steam increases, it soon becomes necessary to set aside or to install a boiler solely for this purpose. Then comes the turning point of the plant. The chances are that it is selling

live steam on an exhaust steam basis, and it soon becomes evident, if the question is investigated, that there is no profit in the business. This condition of affairs has occurred repeatedly and in many cases has both cut down the net earnings of the company and given the heating business a bad name.

The lower the load factor of the plant, the less the inducement to undertake the heating business. In general, a railway plant is better adapted to care for district heating with its exhaust than is a simple lighting plant. As railway plants are generally located on the outskirts or in the country, the heating business may be considered as belonging more especially to lighting companies, and must usually be cared for by them.

While it is true that the demand for heat comes at about the same time of the year as does the demand for light, yet it is also true that it may not come at the same time of the day that the plant is producing its maximum amount of exhaust steam; in other words, the lighting peak and the heating peak may not be coincident, and, in fact, there is no reason why they should be.

So long as the demand does not exceed the supply of exhaust steam, but follows it closely, and the rates are kept up, the plant is in a fair way to make money, but any plant that undertakes to sell live steam on an exhaust steam basis is doomed to failure. . . .

Following is a summary of the information collected; it is divided into three groups; data are from metered customers:

TABLE XLIII.

RECORD OF STEAM REQUIREMENTS IN CENTRAL HEATING.

Class of Service.	Condensation per sq. ft. of radiation per day. Av.	Max.	Min.	Condensation per 1,000 cu. ft. of space heated per day. Av.	Max.	Min.	Ratio cu. ft. space to sq. ft. radiation.	Av. Temp. F.
Residences	3.83	5.20	1.79	51.3	75.2	27.4	76	34°
Residences	2.49	2.49	2.49	34.6	34.6	34.6	72	44°
Residences	1.69	3.22	0.54	....	....	....	...	...
Stores	4.48	10.1	1.01	29.0	50.2	11.4	115	34°
Stores	2.61	4.03	1.18	17.1	37.0	9.2	181	44°
Stores	1.94	3.77	0.64	....	....	....	...	...
Stores	2.32	3.4	1.8	20.0	28.5	13.6	136	40.5°
Offices	5.02	7.94	2.13	51.5	78.0	24.9	81	34°
Offices	2.92	3.70	1.73	25.1	32.2	19.7	120	44°
Offices	1.56	1.83	1.19	26.0	30.4	19.8	60	43.5°
Offices	2.27	3.62	0.89	....	....	....	...	...
Offices	2.97	3.8	1.8	29.0	49.7	14.9	111	40.5°

NOTE.—While there is a large variation between maximum and minimum, it was found when making the tabulation that most of the figures ran very close to the averages given above.

CENTRAL POWER AND HEATING PLANT FOR GOVERNMENT BUILDINGS.*

This report deals with the problem of a central station for thirteen existing and projected government buildings on the Mall and in the vicinity of the White House. The report states: An examination of the several items of unavoidable costs, as also those of possible profit, in the erection and operation of a central plant, led Professor Woodbridge to decide that in order to secure the largest profit, the station should furnish the largest service.

The economies possible under any scheme for installing and operating either a power or a heating central station plant, and the retention of the other in the buildings, would be so seriously narrowed as to make the grounds for advocating such a plant far less tenable than are those found to exist in support of a central plant which shall unite the two divisions of the work. Any such partial treatment of the problem would result, in Professor Woodbridge's opinion, in a costly increase of steam generating plants, including emergency reserves, however the division might be made; in large increased cost in generators and other equipment, if these are segregated, and in an unreduced pay roll under any divided arrangement. The losses which necessarily would be entailed are those due to failure to secure from one division of the service the economic advantages it would contribute to the other, and also those due to the absence of the effective organization and vigilant surveillance which are far more easily obtainable in an aggregated than in a segregated plant. Little, if any, usable space would be gained within buildings; the nuisance of dirt and smoke and heat would be but little abated; and the high cost of smokeless coal would make against economy, whatever the division, and the waste attending low efficiency in engine work would be continued and increased by an extension of the system, and the heat of engine-exhaust steam would be but partially utilized, or wholly lost, if the segregation were on the side of power. . . .

*** Extracts as reprinted in the *Engineering Record*, Feb. 11, 1905, from a report by Professor S. H. Woodbridge, dated Jan. 7, 1905, transmitted to Congress through Bernard R. Green, Superintendent Library Buildings and Grounds.**

The required maximum power and heating capacities of a central station were computed from data furnished from records and logs obtained from the superintendents, custodians, or engineers of those buldings by means of season aggregates, and the daily average.

The power of the proposed combined plant for the above group of buildings is made less than that of the total equipment for segregated plants in the same buildings; first, because of the higher steaming efficiency practically obtainable in the boilers of a highly organized and skillfully operated power plant; and second, for the reason that the isolated plants must each be given a generous factor of safety for performing occasional exceptionally heavy duty as well as providing against breakdown and other emergencies. The serving of the executive group of buildings from one common plant would remove the necessity of furnishing a surplus of power and equipment equal to performing such duties or to meeting such emergencies in all of the buildings at one and the same time. Because of that fact, a substantial reduction in the steam generating plant of a central station may be safely made, Professor Woodbridge states. In place of the present 73 boilers in twenty or more isolated stations, aggregating some 4,500 horse-power, a central station equipment, which shall include boilers for the three additional and large buildings, may not contain more than ten boilers of 400 horse-power each, all in one boiler room. On an emergency, high efficiency boilers having a nominal capacity of 4,000 horse-power could be easily forced to a 6,000 horse-power steam output. . . .

The usual and required emergency equipment for the aggregated plant should have reference to excessive loads of short duration, or to temporary disablement of apparatus which may be quickly restored to service. These occasions must be provided for by reserve or emergency equipment, if the performance of the plant is to satisfy a reasonable demand. In the matter of light and power, that equipment may be either in boilers, engines, and generators, to create and furnish emergency power or demand, or else, with a more equal work of boilers and engines in the storing of electrical energy unused during the long periods of light load, and made available for use in the shorter "peak"

periods of high loads or at times of accident or other emergency. The cost of emergency or accident equipment by the reservoir method is approximately the same as that for the generating method of equal capacity, and the cost of maintenance is somewhat greater in the former than in the latter. The economic gain in operating costs by storage methods is chiefly on the side of reduced fuel consumption. Roughly the capacity of a plant for generating emergency or "peak" power on demand must, in this instance, be approximately 40 per cent. in excess of that of a plant capable of the same work by a stored electrical supply. The usual practice of continuously maintaining full steam pressure in all power boilers, even in summer time, in order to be prepared for a suddenly darkened sky or other emergency, is so productive of fuel waste and of service expense as to warrant the installation and use of a storage plant which shall reduce by 40 per cent. the capacity of the generating equipment, lessen the summer coal consumption, and curtail the expenses of labor service by making night attendants for boilers and engines unnecessary during the summer months. To the advantages named must be added the further important function of the storage battery already noticed, viz., that of evening the work of boilers and of fires and of reducing and removing the causes which tend to produce smoke.

The hourly maximum heat required for warming and ventilating the executive group of buildings during the periods of extreme weather is considerably reduced, Professor Woodbridge points out, by the massiveness and the high specific heat of which the buildings are constructed. Such material in such mass in itself stores large heat quantities, which it partially yields to contained air whenever the temperature of that air falls even a little below the temperature of the material. Such storing and yielding of heat in buildings of monumental character make unnecessary that provision in the power of the heating system which is requisite in lightly constructed buildings for maintaining an even internal temperature during short and sharp depressions of outside temperature. . . .

Weather temperature changes take place with such relatively gradual movement and with such premonitory warnings, that fires can easily be built, forced, banked, or drawn to meet requirements,

as cannot be done on either the electric side in lighting, to meet the quick darkening of the sky or winter daily calls, morning and afternoon, for light; or in office fan work with the dying out of summer breezes, or the quick rise of oppressive humidity or the shifting of heating sunshine from one already heated side of the building to heat the other; or, on the power side, to meet the peak elevator work, as four times a day thousands crowd the elevator service. . . .

Low pressure live steam for heat transmission is economically disadvantageous on account of the larger costs of piping material, of trenches, and of insulation, and because of the large loss of heat in exhaust steam, which, under such conditions, would probably be thrown to waste. High pressure steam service is superior to low only on the score of lower first cost of pipe installation, and a somewhat lower rate of heat loss, the small pipe surface more than offsetting its greater per-unit-of-service rate of heat loss.

The heat quantities lost by steam piping when properly protected from ground and atmospheric moisture, and when thoroughly insulated with the best non-conducting material, is shown in the table for pipes of equal length and for the various sizes indicated, and when supplied with steam at the initial pressure, the steam dropping to atmospheric pressure at the delivery ends. The heat losses are expressed in thermal units per hour, and are probably the lowest economically obtainable by the best practicable protection and insulation, as hereinafter described. The length of pipe is one-fourth mile; the steam quantity passing into pipes for transmission is 150 pounds per minute, a pressure dropping from the initial given to atmospheric at the delivery ends.

TABLE XLIV.

HEAT LOSSES IN ONE-FOURTH MILE OF PROTECTED STEAM PIPE.

Size.	Initial pressure, lbs. to square in.	Heat losses per Hr. B. T. U.
4-inch pipe	45.00	81,600
5-inch pipe	25.00	94,200
6-inch pipe	10.80	101,900
7-inch pipe	6.00	113,300
8-inch pipe	3.70	126,000
9-inch pipe	1.50	138,100
10-inch pipe	1.00	152,500

The table makes evident the economic advantage of high pressure over low in the single matter of relative heat wastes from small and large pipes and is in itself an argument against the use of low steam pressures in large distributing pipes.

The cost of maintenance of high pressure pipe lines would, however, prove larger than that for low pressure pipes, because of expansion strains, and also on account of the higher temperatures and humidities of trench spaces, which would tend to a deleterious condensation of moisture on and to a rusting of the cooler return piping, both of which processes would be more active than if pressure and temperatures were low. . . .

Water may be driven through pipes at any advantageous speed by centrifugal pumps and at a negligible cost, since all the heat of the steam used in driving of pumps to impel water motion may be given to the water moved, and the energy of water motion itself is finally transformed into heat within the water. Furthermore, the heat yielded or conveyed by a cubic foot of water when its temperature is dropped through 30 degrees F. is largely in excess of that yielded or conveyed by a cubic foot of steam, even at very high pressure.

Water pipes have one distinctive and important advantage over steam pipes, since the pressures to which the water pipes are subjected are nearly constant, and for the larger pipes those pressures are relatively low. Strains are therefore light and but slightly variable.

In the matter of durability, under favorable conditions water piping may outlast steam piping. If the system is reasonably tight and the same body of water is continuously used the free oxygen in the water supply having been once expelled or exhausted, internal corrosion is arrested, and the piping will then last indefinitely. The temperature of supply and return mains are so nearly the same that condensation on the external surface of the cooler pipe is retarded, if not prevented, and external corrosion due to that cause is correspondingly prevented and a long life of the piping is equally insured. Furthermore, the temperature of the pipe line is low and the heat loss is proportionately reduced. The necessity of doubling the pipe surface, by running both supply and return pipes of the same size, brings up the total loss for the same

heat transmission to nearly that of high pressure steam piping and of its condensation return piping.

The accompanying table gives the heat losses in thermal units per hour from single water pipes of the sizes given, one-fourth mile long, and containing water at a temperature of 150 degrees, and when the pipes are in parallel, and insulated and protected against moisture in the very best manner practicable, the mean temperature of earth about the trench being 50 degrees F.

TABLE XLV.

HEAT LOSSES IN ONE-FOURTH MILE OF HOT WATER PIPE.

Pipe sizes.	Temperature.	Water Heat Losses B. T. U., per hr.
4-inch	150°	48,100
5-inch	150°	57,200
7-inch	150°	65,800
8-inch	150°	74,400
9-inch	150°	83,300
10-inch	150°	93,000

The use of water makes continuity in gradient lines unnecessary, avoids the use of trench traps, and reduces the number of expansion joints required, so favoring minimum costs in construction, in waste, and in maintenance. Furthermore, some of the largest buildings are at present equipped with hot water heating systems, making the cost of change to adapt them to a central plant relativeiy small. For these reasons of major and minor economies the preference is therefore given to water as the most appropriate vehicle for heat conveyance.

The pressures required to distribute water through a pipe system one-fourth mile in length and in sufficient quantity when cooled 30 degrees to convey per minute the heat equivalent to 150 pounds of steam at atmospheric pressure and condensed to 212 degrees, are shown in the following table:

TABLE XLVI.

PRESSURE REQUIRED TO DISTRIBUTE WATER THROUGH ONE-FORTH MILE OF PIPE.

Pipe sizes.	Steam.*	Water — Volume, gal.	Water — Pressure — Water, column ft.	Water — Pressure — lbs. per sq. in.
4-inch	45.0	606	361.0	157.00
5-inch	25.0	606	112.0	47.00
6-inch	10.8	606	44.7	18.70
7-inch	6.0	606	25.0	10.50
8-inch	3.7	606	10.1	4.20
9-inch	1.5	606	5.95	2.50
10-inch	1.0	606	3.30	1.38

* Initial pressure per square inch.

The two essentials to an economical transmission of heat through long pipe-line distances are: First, such rapidity of travel of the thermal vehicle through the channels that time for leakage of heat from each unit of volume or of weight of that vehicle shall be short: second, such insulation of the pipe surfaces that the rate of heat leakage from each unit of pipe surface shall be small. Swift flow and small leakage are both favored by a use of the smallest pipes practicable. Such pipes have already been noted as possessing other economic advantages.

Defectiveness of insulating material and in methods of its application are responsible for the excessive losses which attend much of present heat distribution from central stations. In scores of cases on record the mean of such loss reaches from 15 per cent. at the lowest to 33 per cent. at the highest of the total heat passing into interred pipe systems. These high losses are partly due, however, to the large number of branch connections made with buildings along the entire lines of pipe lines, and also to a probably inferior manner in insulating and protecting such branches against heat loss.

For the purpose of reducing the first cost of installation and also the after cost of heat loss from pipes, the pipe plan is proportioned for a working pressure difference between the feed and discharge sides of the centrifugal pump of 200 feet hydraulic head, or 100 feet propulsion and 100 feet suction. Such a drop of pressure on the suction side of the pump is made practicable because of the hydrostatic head of 122 feet to which the circulating system would be subjected, due to the height of the fourth floor of the State, War and Navy Buildings. Such static pressure would utilize the suction effect of the pump, and with pump working on the cooler side of the heater, would prevent any liability of a vapor break in the water.

It is further advised that the water be circulated in such manner that in the coldest weather it shall lose not more than 30 degrees in temperature in complete circuit, and the ratio of the length of time the water may be retained in the buildings to that it shall be in traversing the pipes shall be made as large as practicable.

DATA ON CENTRAL STATION HEATING.*

A valuable lot of information concerning the proportions and operation of central station heating is contained in Bulletin 373 of the United States Geological Survey. The pamphlet is a discussion of smokeless combustion in boiler plants, written by D. T. Randall and H. W. Weeks, but the last ten or fifteen pages are devoted to central station heating on account of its relation to smoke abatement in substituting a central heating station for numerous individual heating plants. . . .

The plants range in size from 300 to 16,000 horse-power, though only 25 per cent. are of 600 horse-power or less. Sixteen of the plants have mechanical stokers. The price of coal ranges from $4.60 per short ton in Montana to 90 cents in Illinois, the average cost from all plants being $2.05 per short ton. Both direct and indirect radiation are used, but by far the greater proportion is direct. The greatest distance to which heat is sent from the station varies considerably, but a reasonable distance seems to be about 4,000 to 5,000 feet.

Payment for the use of steam is made in two ways: (1) At a flat rate, based on square feet of radiating surface installed or on 1,000 cubic feet of contents heated, or (2) at a meter rate, based on 1,000 pounds of condensed steam. The price paid per square foot of radiating surface averages 33⅓ cents and varied from 22½ cents to 65 cents. The plants selling on a basis of 1,000 cubic feet of contents charge an average of $4.46, the price varying from $2 to $6. On the basis of 1,000 pounds of condensed steam the payments average 50½ cents, ranging from 40 cents to 66 cents. One plant that sold heat on this basis of 40 cents intimated that such a rate was not profitable.

The hot water plants sold heat only on a basis of square feet of radiating surface installed, the average rate being 17½ cents, and the range from 12½ cents to 25 cents per square foot. Two plants, one selling at 12½ cents and the other at 15½ cents, claimed that their prices were too low for successful operation.

A comparison of the prices charged by central stations, as

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* Extract from an article published in *The Metal Worker*, Sept. 4, 1909.

compared with the cost of fuel only for a house heating boiler, as published in bulletin 366, shows that in many cases the cost of producing heat on the premises equals the price charged by the central station. When heat is purchased the customer avoids the annoyance of having to supervise the operation of the heating plant, as well as the dust resulting from the delivery of fuel and the removal of ashes. Some allowance should also be made for the space that would be occupied by the heater and for the expense necessary to install and keep a boiler in repair.

The following suggestions have been made by the managers of the plants and are worthy of consideration:

Heat from a central plant should be, as largely as possible, a secondary product.

Heating mains should be concentrated and should not extend too far from the station.

Direct radiation should be installed.

Mains should be of sufficient size to avoid the necessity of high pressure at the station.

Heat should be under automatic control.

The flat rate is not a successful basis for payment; the service should be metered.

The costs for coal are for short tons. Many of the plants supplied have very little indirect radiation, and many of them none at all.

EXHAUST STEAM FOR HEATING PURPOSES.*

Amount of Heat in the Exhaust Steam.

"The number of British thermal units exhausted in the steam in any case is equal to the total heat in the steam at the admission to the engine, minus the heat radiated from the cylinder of the engine and also minus the amount of heat absorbed in the actual work done in the steam engine cylinder. Professor Hoffman investigated four different cases.

Case I comprehended a boiler pressure of 100 pounds gauge; pressure at the cylinder 97 pounds gauge; quality of steam at

* Notes based on a paper by Professor J. D. Hoffman, read before the National District Heating Association, Toledo.—*Metal Worker*, June 4, 1910.

cylinder 98 per cent.; steam consumption 34 pounds per indicated horse-power per hour; 1 per cent. loss in radiation from the cylinder, and exhaust pressure 2 pounds gauge. The total heat in the steam is of course found by multiplying the latent heat at the boiler pressure by the quality of the steam; that is, taking into account such moisture as may be in entrainment in the steam and adding to this product the heat of the liquid; that is, the number of heat units required to raise the water from some standard, like 32 degrees to the boiling point. In Case 1, multiplying the total heat of the steam by 34 gives the total supply of heat available to the engine of 39,793 B. T. U. Taking into account that 1 per cent. of the heat is radiated from the engine cylinder, leaves 39,395 B. T. U. available for useful work. One horse-power mechanical work maintained one hour is equivalent to 2,545 B. T. U., so that the amount of heat given into the exhaust per horse-power per hour is 36,850 B. T. U. As compared with the total amount of heat in steam at 2 pounds' pressure the heat given to the exhaust from the steam engine is only 94 per cent. of the amount of heat in the saturated steam at 2 pounds' gauge.

For the second case he assumed an engine receiving steam at 125 pounds' gauge boiler pressure, with the steam at 122 pounds at the engine. The steam consumption was taken at 22 pounds per indicated horse-power per hour; in other respects the data were the same. In this case the total amount of heat delivered to the engine with 2 per cent. reduction for radiation losses amounts to 25,321 B. T. U., so that subtracting 2,545 B. T. U. the equivalent of the mechanical work of 1 horse-power leaves 22,776 B. T. U. given up to the exhaust. This is but 90 per cent. of the amount of heat which is available in the saturated steam at 2 pounds' gauge.

For the third case he assumed the conditions the same as in Case 1, with the exception that the steam was superheated to 150 degrees of superheat. This increased the amount of heat supplied to the engine by the fact that for every one of the 34 pounds required per horse-power per hour, the 150 degrees superheat represent very nearly 75 B. T. U. per pound. The total figures out at 42,411 B. T. U. and with a subtraction for the equivalent of 1 horse-power leaves 39,867 B. T. U. available in the exhaust steam.

This figures out as 102 per cent. of the amount of heat available in the saturated steam at 2 pounds' gauge.

From Cases 1 and 2 it would appear that the greatest amount of heat that can be expected from engine exhausts, for use in heating systems, at or near the pressure of the atmosphere, is 90 to 94 per cent. of that of saturated steam at the same pressure. The percentage will, in most cases, drop much below this value. All things considered, exhaust steam having 80 per cent. of the vaue of saturated steam at the same pressure is probably the safest rating when calculating the amount of radiation to be supplied by the engines. In many cases no doubt this could be exceeded, but it is always best to take a safe value."

EXPLANATION OF HIGH VALUE OF EXHAUST STEAM.

"In plants where the exhaust steam is used for heating purposes and where the amount supplied by direct-acting steam pumps is large compared with that supplied by the power units, it is possible to have the quality of the exhaust steam fairly high. This condition is sometimes met with in practice, and may be the explanation of the statement sometimes heard, that exhaust steam gives better service for heating purposes than saturated steam at the same pressure. It should be understood that saturated steam at any stated pressure always has the same number of British thermal units in it no matter whether it is taken directly from the boiler or from the engine exhaust. A pound of the mixture of steam and entrained water, taken from engine exhausts, should not be considered as a pound of steam. If we are speaking of a pound of exhaust steam without the entrained water as compared with a pound of saturated steam at the same pressure, they are the same, but a pound of engine exhaust of mixture is a different thing. In cases where engine and pump exhausts are mixed, the available heat per pound of steam will be somewhere between 800 and 1,000 B. T. U. As suggested above, however, a safe value for general work is the lower figure."

EXHAUST STEAM HEATING.*

The steam engine of to-day has reached a degree of perfection beyond which it seems improbable that any material advance in efficiency will be made. . . . Under the best conditions possible, a maximum of only 20 per cent. of the heat delivered to the engine in the form of steam is transformed from thermal to mechanical energy, and under commercial conditions of variable load the efficiency seldom exceeds 10 per cent. . . .

The advisability of using exhaust steam as a heating agent has long been recognized, yet there are many who have the false impression that the steam that has passed through the engine possesses but little heat compared with what it had when it entered the cylinder. But steam manufactured at high pressures and reduced to a heating pressure is, in fact, drier, and thus more effective than if sent out at heating pressure. Steam exhausted from the engine answers this condition and is thus in the line of economical duty. This misapprehension should be entirely cleared away by reference to the eminently satisfactory results as obtained by the companies using exhaust steam for heating purposes, of which there are more than one hundred. . . .

The conditions of operation that are necessary, in order that exhaust steam may be utilized in a heating system, vary through wide limits, depending upon the local conditions, class of engines operated, maximum and minimum demand for heat, and the maximum and minimum output of the plant in electrical or power load during the heating and non-heating season. . . .

Competent engineers assert that during the hours when the electric plant is idle and the fires are banked, it costs, approximately, 10 per cent. of what the fuel and labor expense would be to run at full working capacity, which, of course, brings no return, but by the addition of a steam heating plant continuous work is given to boilers and men, and a profit secured in place of this loss. . . .

*** Extract from American District Steam Company's 1905 Catalog, on District Steam Heating.**

HEATING VERSUS CONDENSING.

By present methods employed by the American District Steam Company, the connections at the engines are so arranged that only a needed number of engines will be operated non-condensing when heat is required and furnish their exhaust for heating purposes; while the balance of them can be operated condensing. . . .

To illustrate the increased value of utilizing the heat in the exhaust steam from the engines instead of condensing it, without going into technical arguments, we instance that companies that are operating non-condensing when heat is needed receive an income from the sale of exhaust steam for commercial heating purposes, during the heating season of from seven to eight months, exceeding in amount the cost of fuel at the station for the entire season required to manufacture light, heat and power, plus interest on the cost of the steam heating plant. No advocate of condensing can even hope to approximate such a result.

Electric companies operated by water power find it a great advantage to construct a direct steam heating plant to operate in conjunction with their electrical business, not only on account of the revenue and profit derived from the steam heating enterprise, but the advantage gained in doing away with competition of isolated plants that are forced to construct a heating plant and, naturally, furnish their own light and power as well. . . .

We now use nothing but full-weight strictly wrought-iron line pipe. The increased cost over merchant standard and steam pipe is more than compensated for by greater durability. As to steel pipe, the irregular distribution of carbon, even though in amount less than 1/10 of 1 per cent. used in its manufacture, positively precludes its use at any price.

In some cases it is desirable to return the water of condensation to the station for re-evaporation on account of its purity and also on account of its action in preventing the formation of scale in the boilers, or because of the high cost of water for steam-making purposes. . . . Steel pipe cannot be used for this purpose on account of the pitting and corrosive action which is immediately set up by the water of condensation, thus rendering the life of the steel pipe extremely short. . . .

As an alternative to the use of wood water pipe for returning condensation, this company is also prepared to install, where desired, a cast-iron return main, properly insulated, in which case pipe is used having threaded ends and with special threaded cast-iron couplings similar in a general way to those used for wrought-iron pipe. With this type of construction, however, expansion and anchorage devices are required.

SOME OF THE FACTORS THAT AFFECT THE COST OF GENERATING AND DISTRIBUTING STEAM FOR HEATING.*

The comparative heat value of fuels, taking into consideration relative furnace efficiency, is approximately as follows:

2,000 lbs. coal containing 10,000 B. T. U.=
3.23 bbls. crude oil or 135.66 gals.=
20,000 cubic feet natural gas.

2,000 lbs. coal containing 11,000 B. T. U.=
3.55 bbls. crude oil or 149.1 gals.=
22,000 cubic feet natural gas.

2,000 lbs. coal containing 12,000 B. T. U.=
3.87 bbls. crude oil or 162.5 gals.=
24,000 cubic feet natural gas.

2,000 lbs. coal containing 13,000 B. T. U.=
4.19 bbls. crude oil or 176.0 gals.=
26,000 cubic feet natural gas.

2,000 lbs. coal containing 14,000 B. T. U.=
4.52 bbls. crude oil or 189.8 gals.=
28,000 cubic feet natural gas.

2,000 lbs. coal containing 15,000 B. T. U.=
4.84 bbls. crude oil or 203.28 gals.=
30,000 cubic feet natural gas.

Clean boilers are also imperative in steam plants. The influence of scale on boiler tubes varies largely with reference to its thickness, ranging in decrease in heat conductivity from 3 per cent. to 19 per cent. when from .02 to .085 thick, but not in proportion to thickness, but being very greatly affected by its char-

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* Extracts from a paper by Charles R. Bishop, read before the National District Heating Association, held at Toledo, Ohio, June 1 to 3, 1910.

acter, such as hard, soft, dense or porous. Water, containing from 15 to 20 grains of calcium carbonate, magnesium carbonate and magnesium chloride per United States gallon is fair; when less, is either good or very good; 8 grains or less being considered very good; over 30 grains is very bad. Where the feed water is bad something should be done to either neutralize or eliminate the impurities, through mechanical or thermal means; using boiler compounds or puifying systems—tube cleaners and frequent blow-off of boiler, or by feed-water heaters.

With a good quality of fuel and water and a generating plant containing economical boilers of proper size, selected with reference at least to load conditions, attention should be given to an economical means of utilizing the greatest percentage of heat units contained in the fuel to be burned, and it is generally considered good practice to equip plants of 1,000 or more horse-power with mechanical stokers, adapted to the kind of fuel burned. Such an installation not only increases efficiency, but decreases the firing cost per ton. In hand-fired plants one fireman can attend to the coal, water and ashes for 200 horse-power, while with good automatic stokers, overhead bunkers and down-pour spouts, he can easily attend to 2,000 horse-power or even 3,000 horse-power. The first cost of mechanical stokers is not the only feature for consideration, and those not adapted to the service, poorly designed or constructed, should be avoided on account of liability of shut-downs and large repair bills. A good type of stoker, properly set, is inexpensive to maintain and exceedingly reliable. . . .

It cannot be denied that marked fuel economy will result from proper feed-water heating. This is accomplished in various ways —principally exhaust steam heaters, flue gas heaters, and live steam heaters. As local conditions largely enter into the factors which must determine the installation of these types of apparatus, no general rule can be given, but it can be stated that the fuel savings can range from less than 1 per cent. to over 20 per cent., against which must be charged interest, maintenance and depreciation upon the cost of heaters, and economizers, and extra cost of stack or forced draft apparatus, with allowances for reduction in boiler heating surfaces thus made possible.

After a consideration of the factors which largely enter into the generation of steam, next comes the question of its utilization.

If the plant in mind is a combination electric generating and steam heating plant, it will first be assumed that it intends to have or does have a market sufficient to utilize the full amount of exhaust steam at such times as it is developing the greatest electrical load, and that its steam demand will at least equal the exhaust during that portion of the heating season between November 1st and April 1st. In such cases the question of engine efficiency is not as serious as engine capacity, up to the point of approximately 65 per cent. of engine capacity, although this figure varies with different companies due to the proportion of summer to winter maximum electric load. Some companies desiring to install a sufficient amount of high efficiency engines or turbines to permit of summer operation under economic generating costs, other companies believing the installation of exhaust turbines for summer operation as advisable. The proper installation is a matter quite easy to work out, but depends very largely upon the size of the plant and its power and load factors.

It is a fact that engines act in a sense as reducing valves between the boilers and the heating system, but while performing such functions, produce mechanical energy without appreciably, at least, lessening heat energy; therefore, were there a heat demand in excess of the amount of steam exhausted by high efficiency engine, the deficiency in steam would have to be made up direct from the boilers, and thus the quantity of steam generated in the boilers would not be less than if it had all passed through the engines in the manufacture of a given quantity of engine horse-power or its equivalent in kilowatt hours. By this statement I do not, however, mean to convey the idea that it is advisable to install or continue to operate engines of very low efficiency. . . .

After being exhausted from the engines, the steam enters a main delivery pipe, having at one point a fitting through which the steam is taken to feed-water heaters and through a balanced back pressure valve to the air, such discharge connection being necessary to relieve the engine from any unnecessary back pressure at such times as the quantity of exhaust exceeds the heating

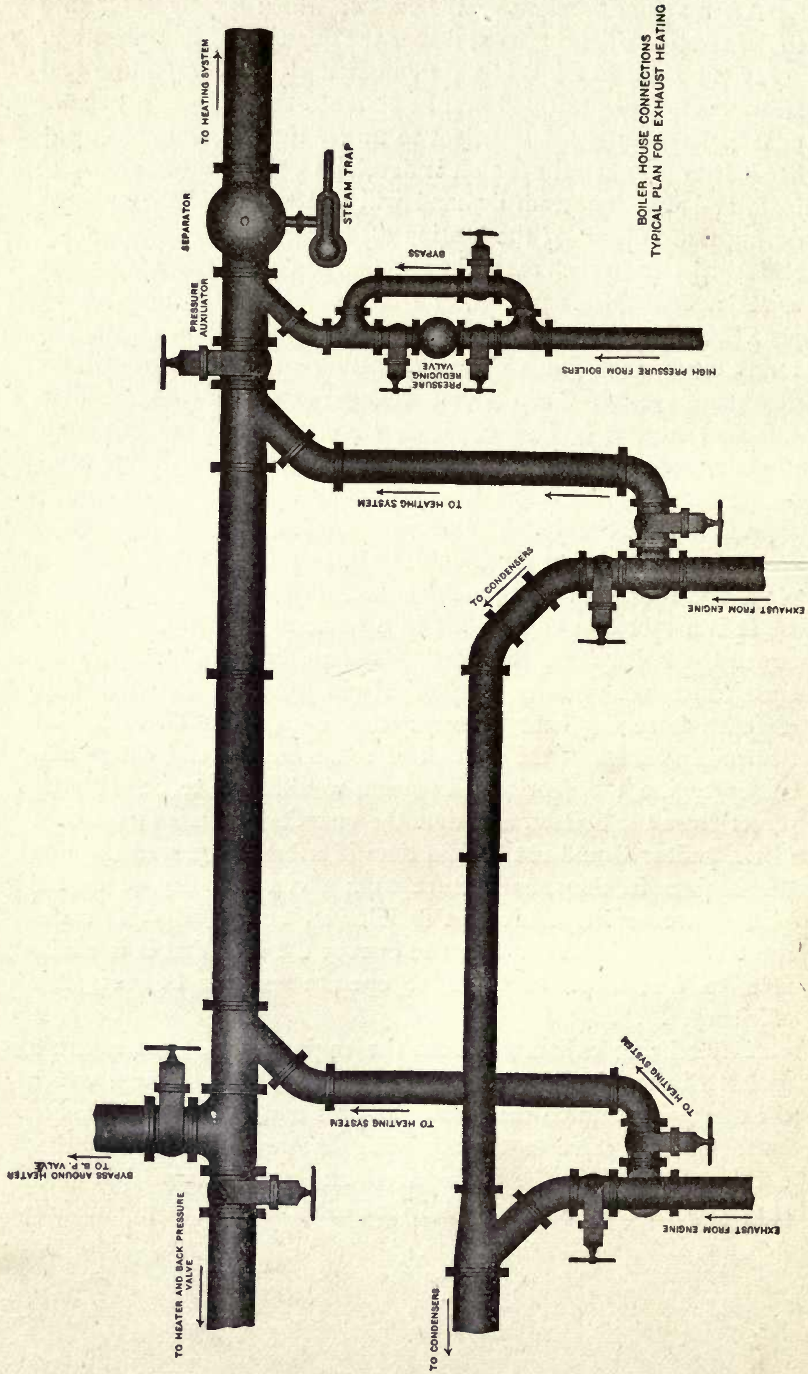

Fig. 56.—Boiler House Connections. Typical Plan for Exhaust Heating.

demand. A direct connection from the boilers is made to the exhaust main for the purpose of supplying the system with the necessary amount of live steam at such times as there exists a heating demand in excess of the exhaust steam supplied. In the large steam connections an automatic pressure reducing valve is placed and in the exhaust steam main, before leaving the station, there is located an oil and water separator, having an automatic trap discharge.

Where the station contains several engines, an additional exhaust main should be installed, so that any engine can be run condensing or exhausting to the atmosphere, while other engines are exhausting into the heating mains. . . . (See Fig. 56.)

Transmission losses are increased or decreased by the following factors:

Efficiency of insulation.

Outside water coming into contact with the distributing mains.

Air in motion coming into contact with the distributing mains.

Actual leakage of steam, through imperfect conditions of the mains, and of expansion joints, or due to drip traps blowing.

Frequent inspection of the distributing mains should be made, and if there is found any indication of leakage, or electrolytic action, immediate steps should be taken to overcome the condition. If outside water is in contact with the steam mains, the insulation, etc., should at once be protected, and even more important, the source of the trouble should be checked, otherwise a recurrence is to be expected, and while the trouble remains the deterioration of the supply system is made rapid.

Street traps should be kept in perfect working condition and they should be inspected at least as often as every seven or ten days.

Normal transmission losses are practically constant in well-insulated and properly installed mains, and are approximately uniform per square foot of mains and fillings and should never exceed .045 pound per hour per square foot. The percentage of loss to output depends upon load factor, and should not exceed 15 per cent. at times of minimum demand during the heating season, nor exceed 1½ per cent. in periods of maximum demand, with an average of not to exceed 4 per cent. during the heating

season. Meters to measure these losses should be in use in all steam plants, and if the losses exceed the amount stated, the cause should at once be determined and overcome. . . .

The real cost of steam delivered to the consumer must necessarily include:

Total generating cost, made up of fuel, water, firing and other station costs.

Transmission losses.

Unaccounted for steam.

Overhead expenses.

Depreciation and amortization.

HEATING SYSTEMS FOR MILLS.*

In the use of direct radiation operated with exhaust steam from non-condensing engines, by placing a light back pressure on the engines to force the exhaust through the pipes at the proper velocity, it should be borne in mind that the adding of one pound of back pressure to an engine cannot be made up by adding one pound to the initial pressure. . . .

Adding back pressure to an engine, especially if it is large and well loaded, should be avoided if possible. From the foregoing it would seem that the entire heat from exhaust steam cannot usually be utilized without running more or less back pressure on the engines. In some instances the circulation of the exhaust is assisted by applying a vacuum at the discharge end of the system, thus doing away with the objectionable back pressure on the engines. This is a decided advantage if the engines are large, but it necessitates the complication of piping, with the cost and running expense of some more or less elaborate system for the maintenance of the vacuum. As a rule, the method of heating by exhaust steam leaves a great deal to be desired and is not advisable unless it is necessary to use non-condensing engines.

Taking low pressure steam from the receiver of a compound engine is a common practice in some mills. If the quantity of steam taken is fairly constant and the engine is designed with that

* Extracts from a paper by A. G. Hosmer, presented before the National Association of Cotton Manufacturers in 1908.

object in view, the results are quite satisfactory, although this idea is usually more popular with the designing than with the operating engineer.

It is well to remember when designing any supply of heat for the purpose of warming buildings of a manufacturing plant that exhaust or receiver steam is available only during running hours, and that the heating system's highest duty is usually required when the engines are shut down; consequently, it is more important to have an efficient service during non-running hours than at times when machinery friction and other causes contribute to the warmth of the workrooms. . . .

A direct system designed for the use of live or boiler steam alone, which is reduced in pressure and sent to the heating pipes, if piping is large and well installed with separate traps for each considerable unit of circulation will be found to be one of the most economical and satisfactory methods of heating. It can be forced or moderated within reasonable limits by raising or lowering the pressure at the reducing valve and, with everything right, will give return for practically all the heat expended.

The benefit of returning the condensation to the boilers is quite often not realized in many plants fitted for so doing, for the reason that a pump designed and used for handling very hot water is one of the most difficult appliances to keep in repair with which the mechanic has to deal. Consequently, the general tendency is to neglect this important feature, and as the heating can be done without it, that is, by running the water to waste, the pump is allowed to stand idle for a great part of the time. This should not be permitted, as the expense of keeping the pump in first-class order is a small item, compared with the loss of the heated condensation.

THE USE OF STEAM FROM THE RECEIVER OF COMPOUND ENGINES FOR HEATING PURPOSES.*

It has been common practice for many years to use the by-products such as exhaust steam and warm water from the steam

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* Extract from paper by Chas. T. Main on "Central Stations vs. Isolated Plants for Textile Mills," presented at a joint meeting of Local Chapters of A. I. M. E. and A. S. M. E., Boston, Feb. 16, 1910.

plant for manufacturing purposes and heating buildings, etc. It has been also very common practice to take steam out of the receiver, between the cylinders of a compound engine for these purposes. In many mills all of the exhaust of simple non-condensing engines is used for manufacturing purposes.

The saving from using the exhaust of a non-condensing engine, which would otherwise go to waste is large, because there is no additional steam required for the engine, unless the back pressure is increased. Any use of the steam is nearly all clear profit and if all of it is used the only part left to charge to power is the difference in B. T. U. due the difference in pressure and the condensation in the engine cylinder and jackets.

There seems to be no good reason why in time the practice of bleeding turbine should not become as common as bleeding engine receivers.

RECEIVER STEAM.

Table XLVII shows the amount of coal chargeable to power when certain percentages of the steam entering the high pressure cylinder are taken out of the cylinder. This table takes into consideration the effects on the economy of the engine of not passing all of the steam into the low pressure cylinder, cylinder condensation, etc.

The percentages in the first column are the percentages of the steam passing the high pressure cylinder which is taken out of the receiver for manufacturing purposes. The second column is the total coal burned and the third is the coal chargeable to power after deducting the coal chargeable to manufacturing.

TABLE XLVII.

EFFECT OF EXHAUST STEAM UTILIZATION ON COMPOUND ENGINE ECONOMY.

Per cent. of exhaust steam used for heating purposes.	Pounds of coal per one horse-power per hour. All coal charged to power.	Net lbs. of coal per one horse-power per hour after deducting for exhaust steam used.
0	1.75	1.75
25	2.06	1.50
50	2.38	1.25
75	2.69	1.00
100	3.00	0.75

If the mill did not obtain its power from steam, so that it could use the low pressure steam of the plant for manufacturing

it would have to maintain a boiler plant of sufficient size to produce an amount of steam equivalent to that bled out of the receiver. The amount of B. T. U. its equivalent in coal chargeable to power is represented by the amount of work done by the engine and the losses due to the presence of the engine. The cost of generating the rest of the steam is chargeable to the manufacturing processes.

HIGH PRESSURE STEAM HEATING.*

In some factory plants, conditions are such that the most economical system to install—both as to first cost and operation—is high pressure steam—that is, a system using steam at 20 to 50 pounds pressure. This condition occurs only where the power load greatly exceeds the heating load and the engines are run condensing, and where calculations show that it would not pay to place a back pressure on the engines and increase the steam consumption; or that it would not be more economical to run with a high vacuum and use live steam in the heating system, varying the pressure by means of the pressure reducing valve to suit the outside temperature. If proper attention is given to such a heating system and the piping is designed to operate with the lowest pressure to be used, satisfactory regulation can be obtained; for steam at 60 pounds pressure has a temperature of 307 degrees and at 20 pounds the temperature is 259 degrees, leaving a considerable range between these extremes. With such a system it is advisable to use coil surface only, as the steam pressure would be too high for the ordinary type of cast-iron radiators.

RELATIVE ECONOMY OF HIGH AND LOW PRESSURE HEATING.

In a paper on "Cost of Heating Storehouses," by H. O. Lacount, presented at the Indianapolis meeting of the A. S. M. E., 1907, it is stated that tests showed that with about 10 pounds steam pressure in the heating system night and day, the average

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* Extract from an article on "The Choice of Heating Equipment for Manufacturing Plants," by G. W. Stanton, *Engineering Magazine.*

temperature throughout the storehouse was substantially the same as when the steam was in the pipes during the daytime only—that is, from 6 A. M. to 6 P. M., but at 60 pounds pressure.

The steam consumption per 24 hours was about 35 per cent. greater with 60 pounds steam pressure during the days than with 10 pounds continuously.

CHAPTER XII.

THE STEAM LOOP.

This interesting method commonly employed for returning live steam drips to the boilers in power plants and also used in low-pressure plants is here described by extracts from two papers or articles, accompanied by illustrations.

THE STEAM LOOP.

This device is well described in a paper by Walter C. Kerr, *Journal of the Franklin Institute,* February, 1891, from which the following extracts are taken:

" Steam loop," a name which, though almost meaningless, seems very consistent with its simplicity. The name has the further merit of not portraying any of its functions or peculiarities, and hence cannot be an embarrassing restraint, as is so frequently the case with names attached to mechanical apparatus.

That so simple an application of Nature's laws as is involved in the steam loop should not have been turned to useful effect earlier is, at first thought, strange, but as one looks deeper into the subject, the reasons become more apparent. While no engineer is unfamiliar with the phenomena on which it depends, it has been interesting to note that even those best informed in practical steam engineering or theoretical research in thermodynamic science, seldom understand its action on first acquaintance, though they soon recognize in it a new combination of functions.

Its mission is the simple and useful one of returning water of condensation to steam boilers. Its chief characteristics are that its action is continuous, rapid and positive, and that it is a closed system operating under widely varying conditions, without valves or adjustments. Its construction is simply that of ordinary piping.

The principles on which its action depends are as follows:

Differences of pressure may be balanced by a water column.

Vapors or liquids tend to flow to the points of lowest pressure.

Rate of flow depends on difference of pressure and mass.

Decrease of static pressure in a steam pipe or chamber is proportional to rate of condensation.

In a steam current water will be carried or swept along rapidly by friction.

To these simple statements there will probably be no dissent. We have all used them in many ways, and some of them have dis-

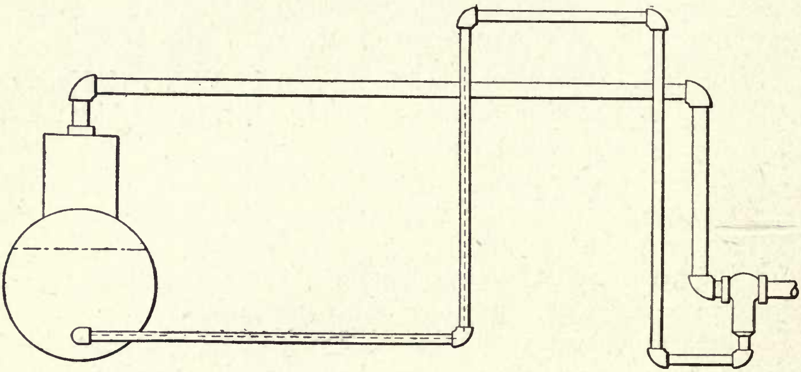

Fig. 57.

agreeably used us in a manner quite unwelcome. But it remained for the steam loop to collect a few of these erratic agents and from them create a useful system, combining the certainty of flow due to difference of pressure, with the quiet uniformity with which steam condenses, and with the force we see uselessly expended in the hammering of our steam heating apparatus.

It will be evident that the steam loop, therefore, contains no mysterious factors, even though, like the steam injector, it has been called a paradox.

Fig. 57 shows a steam pipe passing from the boiler to a separator near the engine, which separates the water of condensation and entrainment from the steam. The drip from the separator is below the boiler, and evidently were a pipe run from this drip outlet directly to the boiler we would not expect the

water to return uphill. Moreover, the pressure in the boiler is (say), 100 pounds, while in the separator it is only ninety-five pounds, due to the decrease in pressure in the steam pipe by reason of which the steam flows to the engine. Thus the water must not only flow up hill to the boiler, but also must overcome the difference in pressure. The device to return it must perform work, and in so doing heat must be lost. The loop, therefore, may be considered as a peculiar motor doing work, the heat expended being radiation from the upper or horizontal portion.

We are now prepared to examine its mechanical operation, which is best done with the model in action. The form of separator is immaterial, there being many kinds, differing more or less in construction and efficiency. The one in model is simply an elbow turned down into the body of the device throwing the steam against a perforated plate, above which the dry steam is removed by a pipe leading to engine, while the water collects below.

From the separator drain leads the pipe called the "riser," which at a suitable hight empties into the "horizontal." This leads to the "drop-leg" connecting to the boiler anywhere under the water line. The riser, horizontal and drop-leg form the loop, and usually consist of pipes varying in size from ¾ inch to 2 inches, and are wholly free from valves, the loop being simply an open pipe giving free communication from separator to boiler. (For convenience, stop and check valves are inserted, but they take no part in the loop's action.) Suppose steam is passing, engine running and separator collecting water. The pressure of 95 pounds at the separator extends (with even further reduction) back through the loop, but in the drop-leg meets a column of water (indicated by the heavy broken line), which has risen from the boiler, where the pressure is 100 pounds, to a hight of about 10 feet. That is to the hydrostatic head equivalent to the 5 pounds difference in pressure. Thus the system is placed in equilibrium.

Now, the steam in the horizontal condenses slightly, lowering the pressure to 94 pounds, and the column in the drop-leg rises 2 feet to balance it, but meanwhile the riser contains a column of

mixed vapor, spray and water, which also tends to rise to supply the horizontal as its steam condenses, and being lighter than the solid water of the drop-leg, it rises much faster. If the contents of the riser have a specific gravity of only one-tenth of that of the water in the drop-leg, the rise will be ten times as rapid, and when the drop-leg column rises 1 foot, the riser column will lift 10 feet. By this process the riser will empty its contents into the horizontal, whence there is a free run to the drop-leg and thence into the boiler. In brief, the above may be summed into the statement that a decrease of pressure in the horizontal produces similar effects on contents of riser and drop-leg, but in degree inversely proportional to their densities. When the condensation in horizontal is maintained at a constant rate sufficient to give the necessary difference in pressure, the drop-leg column reaches a hight corresponding to this constant difference, and rises no further. Thus, the loop is in full action, and will maintain circulation so long as steam is on the system, and the differences of pressure and quantities of water are within the range for which the loop is constructed. . . .

Generally speaking, the limits within which the steam loop is applicable are very wide, for the principle applies quite as well to great as to small differences of pressure. Similarly, an enormous quantity of water may be handled quite as easily as a small amount. The action will continue reliably through long pipes, overhead or underground. Water may be lifted from levels far below the boilers. The use to which the steam may be applied after the loop and separator have dried it, of course has no effect upon the loop system. Wherever steam is so used that it condenses rapidly, as in dryers, steam heating systems, jacket steam kettles, etc., the loop can be applied to the return of this water of condensation the same as from an ordinary separator, and that, too, against any difference of pressure.

The above statements are made to illustrate how thoroughly and completely the loop can be applied to a wide range of conditions, but when we come down to the practical application, and say how far is it expedient to apply it, the field contracts somewhat.

The loop's application is limited most often by the head room

for its erection. If the pressure in a separator, dryer, or return from a steam heating system be 10 pounds below the boiler and a loop about 30 feet high is necessary to make the return, it is evident that a difference of 50 pounds in pressure requires a loop about 150 feet high, and the riser, drop-leg and large portion of the horizontal being well covered with non-conductor, such a loop would operate efficiently, but generally speaking, a line of small pipe of that hight would seldom be convenient, inasmuch as it would require some peculiar structure to hold it, or possibly might need to be erected on the side of a smokestack. In high city buildings such a loop may be practicable where convenient air shafts allow easy support, but in ordinary manufacturing plants it would seldom be constructed.

While speaking of difference of pressure, attention should be called to the fact that the absolute pressure is of no importance, as a loop will work quite as well under low pressure as high. Its construction and operation recognize only the difference of pressure. A special case occurs, however, where the difference of pressure is very large compared with the lowest pressure in a system. For instance, if a boiler carries 25 pounds of steam and at the end of a series of heating or drying coils the pressure is 1 pound, then with a loop about 100 feet high it would be evident that if the condensation in the horizontal were so performed as to even produce a perfect vacuum, the water column in drop-leg would stand about 80 feet high, but it is doubtful whether the pressure of 1 pound at foot of riser plus the 14.7 pounds due to vacuum would be sufficient to force the contents of riser up 100 feet and into the horizontal. It certainly would not be sufficient if there were a considerable quantity of water to be handled, thus causing a high specific gravity of riser contents. Such case, however, is so seldom met in practical loop application, that it scarcely need be considered a limiting condition.

Roughly speaking, differences of 10 to 15 pounds are the largest experienced in good practice, and the loop can generally be conveniently erected to operate against such differences, and where excessive discrepancies in pressure are observed, it is usually very desirable to make such changes as will diminish differences, they being usually due to faulty piping. While, there-

fore, excessive difference of pressure is practically a limiting condition to steam loop practice, it is not found to be an annoying interference. . . .

Throughout this discourse the description has been confined almost wholly to the application of the loop to the one case where moisture is to be removed before steam passes to an engine or pump. It is thought best to keep this one case clearly in mind, for the loop thoroughly understood on this basis may be easily conceived to serve similar purpose in any other connection. Where live steam is used for drying purposes, the loop may be

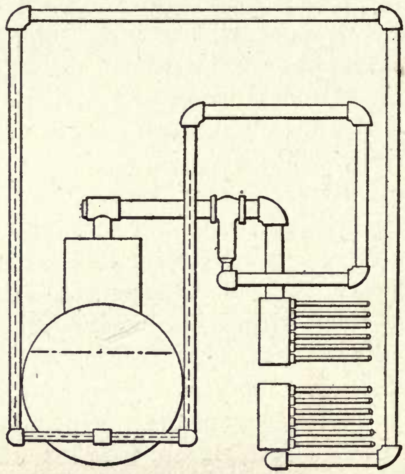

Fig. 58.

attached directly to the return, as shown in Fig. 58, thus maintaining a powerful circulation through the heating coils and ridding the system from the condensation which is the natural product of the heating or drying process. In this service, however, the loop has opened up a new feature, that of drying the steam before it enters such heaters, and it is found to yield very beneficial results, by keeping up temperatures and pressures. Similarly, steam kettles, jackets of steam jacketed cylinders, and even steam heating apparatus can be handled with ease and efficiency. Much apparatus of this nature, however, is throttled down to a degree that seriously interferes with loop application, and in ordinary steam

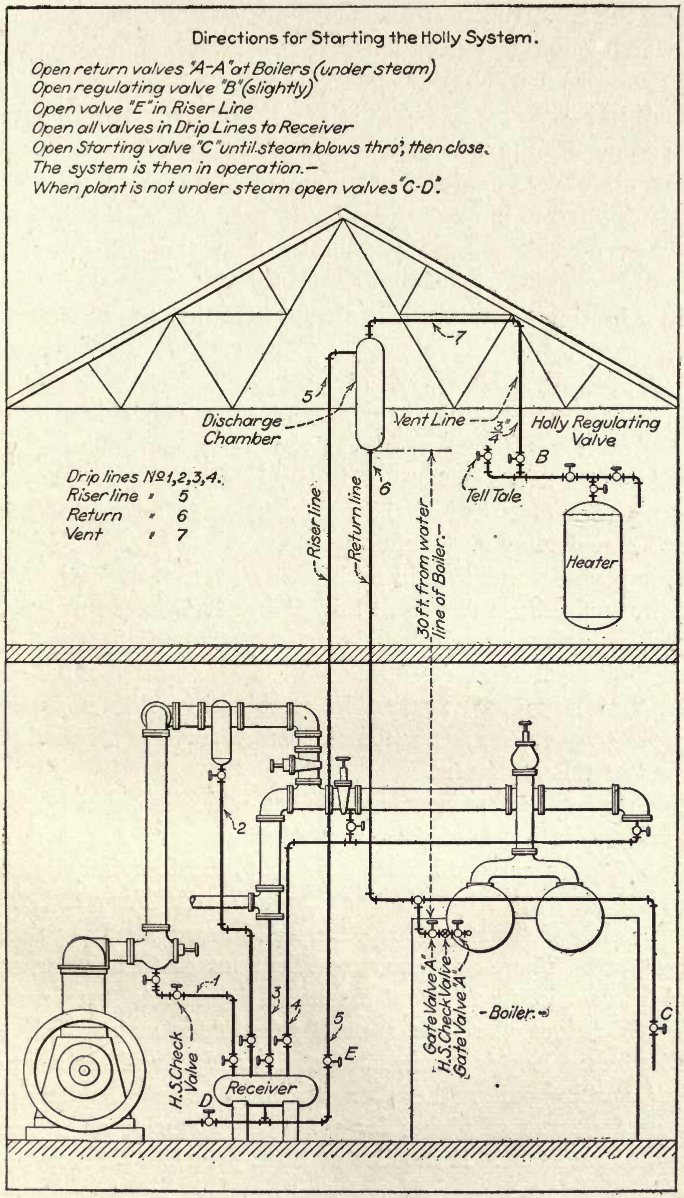
Directions for Starting the Holly System.
Open return valves "A-A" at Boilers (under steam)
Open regulating valve "B" (slightly)
Open valve "E" in Riser Line
Open all valves in Drip Lines to Receiver
Open Starting valve "C" until steam blows thro'; then close.
The system is then in operation.–
When plant is not under steam open valves "C-D".
7
5
Discharge Chamber
Vent Line
3/4"
Holly Regulating Valve
B
6
Tell Tale
Drip lines No 1,2,3,4.
Riser line " 5
Return " 6
Vent " 7
Riser line
Return line
30 ft. from water line of Boiler.–
Heater
2
1
3
4
5
E
H.S. Check Valve
Receiver
D
Gate Valve "A"
H.S. Check Valve
Gate Valve "A"
Boiler
C

Fig. 59.

heating the opportunity is exceptionally good for large air leakage, which would be deleterious. It is, therefore, not easy to concisely state to just what purposes the loop may be practically applied, but it is safe to say that it is desirable on any live steam pipe or any high pressure or unthrottled dryer, heater or jacket.

Fig. 59 shows a modern arrangement of the Holly return system as applied to the draining of high pressure steam mains and branches,

THE STEAM LOOP.*

In a heating apparatus it is always best that the condensation be returned by gravity alone, and this can usually be effected without difficulty. However, conditions are occasionally met with that tax the ingenuity of the engineer, and render it necessary to employ mechanical means, but as the water can always be returned by one means or another, it may be said that, at the present time, no plant that permits the condensation to be wasted can be termed complete or successful.

It is perhaps true that in no branch of steam engineering has more ingenuity been displayed than is involved in the production of appliances for returning condensation from radiation and other condensing mediums located below the water line in a boiler.

An ordinary duplex pump in combination with a receiving tank, pump, governor and automatic return trap are all good illustrations of this statement.

We also have a more recent device which is perhaps not so well known, but it is simple, and when properly installed, successful. This appliance is the steam loop, the foundation patents of which are owned by Westinghouse, Church, Kerr & Co. In the operation of the steam loop, the well-known vacuum principle is employed. There are many modifications of this device, some of which are simply to avoid patent infringements, but, in all of the modifications, the same general principle is maintained.

The steam loop is constructed of iron pipe, and has no movable or working parts, other than what may be found on any ordinary arrangement of piping for conveying steam to an engine or system of radiators. It consists of three principal parts arranged

* These extracts are from a paper by James Mackay, published several years ago (date uncertain), in *Domestic Engineering.*

something like an ordinary syphon, the long leg being termed the main riser, the short leg the drop return, and the horizontal, connecting the two legs together, is termed the condenser. A refer-

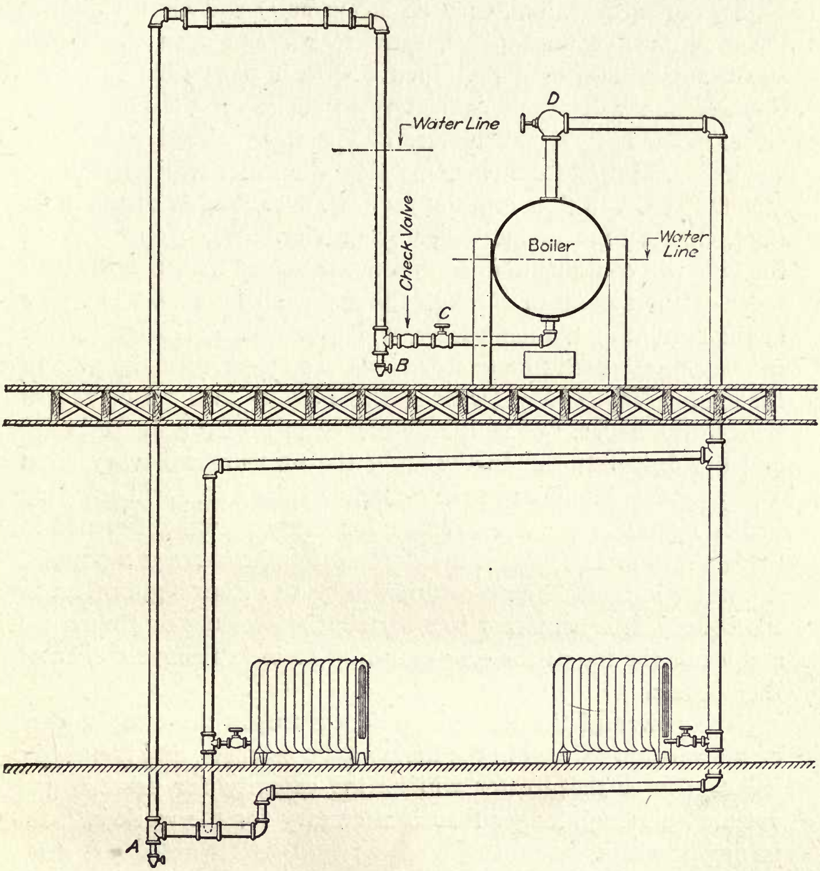

Fig. 60.

ence to the accompanying diagram, Fig. 60, will convey an idea as to the operation of this device. The main riser and drop return are of 1¼-inch or 1½-inch pipe; the horizontal condenser of 2½ or 3-inch pipe, connected with the main riser by an eccentric reducer with the eccentric at the top, and with the drop return

by a similar reducer with the eccentric at the bottom. A compression air valve or pet cock is placed at each end of the loop at A and B in the diagram; a gate and check valve at C upon the return just before it enters the boiler.

To put in operation, turn on steam at D and close valve C. Open A until steam appears, then close it and open B. When steam appears, close B and open C. In a short time sufficient condensation will accumulate at A to choke off the passage of steam to the horizontal condenser. The steam already there will condense and produce a vacuum; then the water which has accumulated at C will flow into the vacuum, over the top of the main riser, thence back to the boiler by gravity. When once started the loop will continue in operation until stopped by some derangement of the system or through the absence of steam. The pipe at the bottom of the main riser, which acts as a receiver for the condensation, should be one or two sizes larger than the pipe in the main riser. The water line will be higher in the drop return than in the boiler, due to the difference in pressure in the boiler and the drop return. In the latter the pressure will vary from 1 to 3 pounds less than in the boiler, and the water will stand 27 inches higher in the drop return for every pound difference in pressure. For instance, if the difference in pressure is 2 pounds, the water line in the drop return will be 54 inches higher than in the boiler. It is necessary to ascertain the pressure on the return and locate the top of the drop return at 10 inches above the water line therein.

The power of the steam loop depends upon the completeness of the vacuum formed in the horizontal condenser, and the longer the condenser the greater will be the capacity for producing a vacuum, although longer than is necessary for the work in hand is simply waste.

The loop will work under all steam pressures, but is not as reliable at low as at high pressure, say from 20 to 50 pounds. Still, good results are being obtained upon low pressure steam heating apparatus at pressures below 5 pounds.

CHAPTER XIII.

NON-CONDUCTING COVERINGS. MISCELLANEOUS TABLES AND FURNACE TESTS.

TESTS OF STEAM PIPE AND BOILER COVERINGS BY C. L. NORTON, OF THE MASSACHUSETTS INSTITUTE OF TECHNOLOGY, FOR THE MUTUAL BOILER INSURANCE CO., 1898.

TABLE XLVIII.

DATA OF PIPE AND BOILER COVERING TESTS.

Name and Maker.	B. T. U. loss per sq. ft. pipe surface per minute.	Per cent. or ratio of loss to loss from bare pipe.	Thickness in inches.	Weight in ounces 4 in. diam., per ft. of length.
Nonpareil Cork Standard—Nonpareil Cork Co......	2.20	15.9	1.00	27
Nonpareil Cork Octagonal—Nonpareil Cork Co.....	2.38	17.2	.80	16
Manville High Pressure—Manville Covering Co....	2.38	17.2	1.25	54
Magnesia—Keasby & Mattison Co...............	2.45	17.7	1.12	35
Imperial Asbestos—H. F. Watson...............	2.49	18.0	1.12	45
"W. B."—H. F. Watson......................	2.62	18.9	1.12	59
Asbestos Air Cell—Asbestos Paper Co...........	2.77	20.0	1.12	35
Manville Infusorial Earth—Manville Covering Co...	2.80	20.2	1.50	..
Manville Low Pressure—Manville Covering Co.....	2.87	20.7	1.25	..
Manville Magnesia Asbestos—Manville Covering Co..	2.88	20.8	1.50	65
Magnabestos—Keasby & Mattison Co.............	2.91	21.0	1.12	48
Moulded Sectional—H. F. Watson...............	3.00	21.7	1.12	41
Asbestos Fire Board—Asbestos Paper Co.........	3.33	24.1	1.12	35
Calcite—Philip Carey Co......................	3.61	26.1	1.12	66
Bare Pipe	13.84	100.	....	..

TABLE XLIX.

PARTIAL COMPONENTS OF DIFFERENT COVERING MATERIAL.

Name.	Percentage Composition	
	Mg.CO3 Carbonate of Magnesia.	Ca.SO4 Sulphate of Calcium.
K. & M. Magnesia................	80 to 90	3
Manville H. P. Lining...........	less than 5	65 to 75
Watson Moulded	20 to 25	50 to 60
Carey Calcite	less than 5	75
Manville Magnesia Asbestos.......	10 to 15	none

TABLE L.

THE SAVING, IN DOLLARS, DUE TO THE USE OF VARIOUS COVERS.

Name.	Loss per sq. ft. B. T. U. at 200 lbs.	Saving B. T. U. per sq. ft.	Saving per year per 100 sq. ft.
Nonpareil Cork Standard	2.20	11.64	$37.80
Nonpareil Cork Octagonal	2.38	11.46	37.20
Manville Sectional High Pressure	2.38	11.46	37.20
Magnesia	2.45	11.39	36.90
Imperial Asbestos	2.49	11.35	36.80
"W. B."	2.62	11.22	36.40
Asbestos Air Cell	2.77	11.07	36.00
Manville Infusorial Earth	2.80	11.04	35.85
Manville Low Pressure	2.87	10.97	35.65
Manville Magnesia Asbestos	2.88	10.96	35.60
Magnabestos	2.91	10.93	35.50
Moulded Sectional	3.00	10.84	35.20
Asbestos Fire Board	3.33	10.51	34.20
Calcite	3.61	10.23	33.24
Bare Pipe	13.84	.....	.....

Generally speaking, a cover saves heat enough to pay for itself in a little less than a year at 310 ten-hour days, and in about four months at 365 twenty-four-hour days.

It is evident that the decision as to the choice of cover must come from other considerations, as well as from the conductivity.

The question of the ability of a pipe cover to withstand the action of heat for a prolonged period without being destroyed or rendered less efficient is of vital importance.

TABLE LI.

LOSS OF HEAT AT 200 POUNDS FROM BARE PIPE.

Condition of Specimen.	B. T. U. lost per sq. ft. per minute.
New pipe	11.96
Fair condition	13.84
Rusty and black	14.20
Cleaned with caustic potash inside and out	13.85
Painted dull white	14.30
Painted glossy white	12.02
Cleaned with potash again	13.84
Coated with cylinder oil	13.90
Painted dull black	14.40
Painted glossy black	12.10

STEAM PIPE COVERING AND ITS RELATION TO STATION ECONOMY.

By H. G. Stott.*

An attempt was made to determine the law governing the effect of increasing the thickness of the insulating material, and for all the 85 per cent. magnesia coverings the efficiency varied directly as the square root of the thickness, but the other materials tested did not follow this simple law closely, each one involving a different constant. . . .

To determine which covering is the most economical for following quantities must be considered.

1. Investment in covering.
2. Cost of coal required to supply lost heat.
3. Five per cent. interest on capital invested in boilers and stokers rendered idle through having to supply lost heat.
4. Guaranteed life of covering.
5. Thickness of covering.

From an inspection of the first three quantities it is apparent that the covering which shows a minimum total cost of the three at the end of a specified time is the best covering to adopt, for the loss in heat at the end of ten years may readily cost over three times as much as the first cost of covering. . . . While pipe covering is a relatively small portion of the many problems confronting engineers, yet its scientific solution will yield rich results out of all proportion to the time required to solve it.

There seems to be no reason for the former practice of putting on different thicknesses of covering on different sized pipes, excepting the mechanical difficulty of applying a very heavy covering to a small pipe. This difficulty can be overcome by putting the covering on in two separate layers, and this plan should be used on all sizes in order that the joints may be broken, as poor joints may reduce the efficiency of the best covering 6 per cent. or more.

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* Extracts from paper read before Twenty-third Convention Association of Edison Illuminating Companies.

TABLE LII.

COMPARISON OF PIPE COVERING TESTS.

Material.	G. M. Brill, June, 1895. 110 pounds.	C. L. Norton, 1898.	W. M. Dickinson, H. & V., Sept., 1894. 80 pounds.	John A., Laird, March, 1895. 25 pounds.	Steam Users' Association, Boston, May, '96. 10 pounds.	Steam Users' Association, Boston, May, '96. 200 pounds.	Geo. H. Barus, 1902. 80 pounds.	Geo. H. Barus, 1902. 150 pounds.	From B. & W. Company.	R. C. Carpenter.	H. C. Stott.	Average.
Bare pipe	100	100	100	100	100	100	100	100	100	100	100	100
Magnesia	14.8	17.7	23	31.0	26.2	28.4	16.5	15.8	21	18.8	15.8	20.8
Rock wool	9.48	...	17	...	39.5	33.3	...	...	...	20.9	...	24.0
Mineral wool	10.52	...	20	...	...	...	...	...	20	19.3	...	17.4
Fire felt	18.56	...	...	34.0	...	...	...	...	...	25.9	...	25.8
Hair felt	15.60	18.9	...	...	...	...	...	...	15	15.0	...	16.1
Fossil meal	32.47	...	...	34.6	...	...	...	...	22	29.7	...	29.7
Asbestos sponge	..	...	22	34.0	...	...	14.6	13.7	...	18.8	20.6	20.6
Asbestos air cell	..	20	...	...	26.1	28.4	24.6	...	...	...	19.0	23.6
Manville sect.	12.92	17.2	...	...	...	...	...	...	...	...	...	15.1
Nonpareil cork	..	15.9	...	...	23.2	25.4	...	15.1	...	...	18.7	19.7
Magnabestos	..	21	...	...	38.3	32.1	...	...	...	...	...	30.5

EXPLANATION.—The condensation in bare pipe is taken as 100 per cent. The figures opposite the different materials show the percentage of steam condensed when the covering is used.

ECONOMIC VALUES OF STEAM PIPE COVERINGS.*

The economic value of non-conducting covering for steam pipes has long been recognized. Especially in cases where the heat radiated from the pipes cannot be utilized, the loss, in a very short time, will more than equal the cost of a good covering. According to Carpenter, the heat losses per square foot of surface for small uncovered pipes is from 375 to 400 B. T. U. per hour. This would mean an annual loss of 30 cents per square foot of such surface, 75 to 80 per cent. of which could be saved by the use of covering. This fact makes the decision to use some kind of pipe covering an easy one, but when the selection of covering is to be decided one immediately encounters much contradictory data.

Reports of tests of pipe coverings are furnished by all dealers, and, strange to say, they always show that the particular brand advertised is superior to all others. This leads up to a brief description of the different classes of covering.

It is generally admitted that loose wool, hair, cotton or feathers are the best non-conductors; but as these materials are all combustible, their use is practically out of the question, except possibly in the case of hair. The materials which are most used at the present time are magnesia, rock wool, mineral wool, fire felt, hair felt, fossil meal, asbestos sponge, nonpareil cork and asbestos paper and air cell. According to Geo. M. Brill (A. S. M. E., Vol. XVI), the composition of various substances is shown by the following table:

TABLE LIII.

COMPOSITION OF COVERINGS (BRILL).

	Magnesia.	Rock wool.	Mineral wool.	Fire felt.	Hair felt.	Fossil meal.
Moisture at 100° C.	1.23	0.00	0.00	0.16	5.55	1.78
Organic matter	7.64	0.00	0.00	11.00	90.85	10.02
Silica	4.05	43.48	48.62	38.18	0.11	85.43
Iron and aluminum oxids	3.74	11.95	9.20	9.56	0.28	1.78
Lime (CaO)	22.95	22.96	24.10	0.10	2.33	0.32
Magnesia (MgO)	19.58	18.24	17.26	37.55	0.21	0.13
Carbonic acid (Co_2)	38.00	0.00	0.00	0.00	3.75	0.00
Sulphurous acid (So_2)	1.57	4.67	1.75	3.60	0.00	0.00

* Extract from article in the *Metal Worker,* August 13, 1904.

It must, of course, be appreciated that the composition must vary, depending upon very many causes, but these results show the probable composition of the different coverings. It is not advisable to use hair felt alone for pipe covering on account of the tendency to char and break away from the pipe. This, however, may be overcome by the use of asbestos paper next the pipe, and on the outside. A very good and comparatively inexpensive covering is made up of several layers of corrugated asbestos paper forming dead air cells to prevent the radiation of heat.

A great deal of data relative to the efficiency of various forms of covering has been published, and the attempt has been made to show comparatively the results of the experiments which have been made by different engineers.

The results given by nine different authorities have been compiled and are given in the accompanying table. This table has been plotted assuming the condensation of an uncovered pipe to be 100 per cent., the various figures given opposite the different materials used for covering showing the percentage of steam which is condensed in the covered pipes. For instance, in the case of magnesia, the average condensation is 20.8 per cent, as much when the pipe is covered with magnesia as it is when a bare pipe is used; in other words, the saving by the use of magnesia covering averages 79.2 per cent.

The results show quite wide variations, but from the whole a much better average may be taken than can be obtained by studying one or two tests independently. An average saving of from 70 to 85 per cent. is shown by the use of covering.

The proper care of covering has a great deal to do with its effectiveness. If from any cause it becomes loose or disconnected, its efficiency is materially affected. It should be thoroughly inspected and repaired every year, the slight cost for this attention being more than covered by the saving effected by greater efficiency. A relatively poor covering properly applied may be made to give better results than a better one which is put on in a slipshod manner.

TABLE LIV.

MINIMUM TEMPERATURE RECORDED IN VARIOUS PARTS OF THE UNITED STATES.

	1886.	1887.	1888.	1889.	1890.	1891.	1892.	1893.	1894.	1895.	Lowest in ten years.
ALA.—Montgomery	15	13	18	21	21	23	20	17	13	8	8
ARIZ. Prescott	−4	8	−12	−8	3	..	..	..	..	..	−12
ARIZ. Tucson	..	..	..	..	..	11	16	22	18	..	11
ARK.—Little Rock	10	0	7	17	16	20	10	11	1	−2	−2
CAL. Los Angeles	36	33	31	32	34	33	35	31	32	34	31
CAL. Sacramento	34	28	19	31	29	26	26	28	26	28	19
COLO.—Denver	−11	−18	−20	−7	−8	−7	−17	−2	−8	−15	−20
CONN.—New Haven	−2	−5	−4	−3	4	3	0	−3	−5	−7	−7
FLA.—Jacksonville	31	22	28	30	27	30	29	24	14	14	14
GA.—Atlanta	8	9	13	14	17	18	13	8	4	0	0
IDAHO Boise City	−7	6	−28	2	−9	..	..	..	..	..	−28
IDAHO Idaho Falls	..	..	..	..	..	..	−22	−22	−28	−32	−32
ILL.—Chicago	−14	−15	−18	−11	−5	−8	−10	−16	−9	−15	−18
IND.—Indianapolis	−11	−12	−6	−1	4	−3	−5	−15	−7	−14	−15
IND. T.—Fort Sill	1	0	−7	7	6	..	..			..	−7
IOWA—Des Moines	−20	−24	−27	−13	−18	−10	−26	−16	−27	−18	−27
KAN.—Dodge City	−18	−17	−18	−8	−6	−0	−11	−7	−15	−14	−18
KY.—Louisville	−5	−5	8	6	13	7	4	−10	−5	−10	−10
LA.—New Orleans	28	21	29	32	30	30	23	29	21	16	16
MASS.—Boston	−2	−5	−6	−1	0	+2	0	−4	−7	−6	−7
MD.—Baltimore	3	7	9	3	12	16	12	1	7	1	1
ME.—Portland	−5	−15	−12	−8	−4	−4	−5	−9	−15	−11	−15
MICH. Detroit	−12	−3	−7	−8	+8	2	−3	−10	−11	−8	−12
MICH. Marquette	−15	−21	−27	−21	−12	−12	−10	−19	−17	−16	−27
MINN.—St Paul	−36	−36	−41	−25	−22	−25	−25	−26	−25	−26	−41
MISS.—Vicksburg	17	10	18	24	24	22	16	20	15	4	4
MO.—St. Louis	−10	−10	−12	0	4	4	−2	−2	−11	−12	−12
MONT.—Helena	−15	−40	−41	−15	−29	−24	−22	−42	−26	−17	−42
N. C.—Charlotte	11	8	16	13	19	19	18	5	2	1	1
NEB.—Omaha	−18	−22	−25	−10	−14	−9	−26	−16	−22	−20	−26
NEV. Carson City	..	..	−10	−7	−22	0	2	8	−7	−4	−22
NEV. Winnemucca	9	−3	−28	−14	−23	−8	3	−19	−11	−14	−28
N. D.—Bismarck	−36	−44	−37	−34	−35	−33	−34	−41	−33	−39	−44
N. H.—Manchester	..	−4	−11	−9	−6	−7	−3	−9	.	..	−11
N. J. Atlantic City	5	−2	2	2	10	14	9	−4	5	..	−4
N. J. New Brunswick	..	..	..	..		..	−1	−10	−1	−10	−10
N. MEX.—Santa Fe	−3	−8	−2	−1	−2	−6	1	5	0	−11	−11
N. Y. Albany	−10	−15	−10	−5	−4	−5	−5	−6	−11	−12	−15
N. Y. New York	0	6	2	2	6	9	8	1	1	−3	−3
OHIO—Columbus	−11	−5	2	1	7	5	−5	−12	−4	−8	−12
OKLA.—Oklahoma City	..	..	..	..	..	10	−11	−2	−8	−8	−11
ORE. Baker City	..	..	..	..	−14	−11	−12	−17	−7	−3	−17
ORE. Portland	17	9	−2	23	10	23	20	8	18	25	−2
PA. Philadelphia	0	8	2	2	9	12	10	0	4	−3	−3
PA. Pittsburgh	−9	4	1	−1	5	9	2	−3	−4	−6	−9
R. I.—Narragansett Pier	..	..	..	..	..	..	−1	−4	−7	−7	−7
S. C.—Charleston	22	17	26	26	25	29	25	20	14	12	12
S. D. Pierre	..	..	..	..	..	−11	−30	−26	−28	−27	−30
S. D. Yankton	−24	−29	−28	−18	−22	−19	−32	−22	..	..	−32
TENN.—Nashville	−2	−2	2	12	16	17	10	3	−2	−6	−6
TEX.—San Antonio	26	17	11	28	21	25	19	26	16	11	11
UTAH—Salt Lake City	5	9	−17	5	−6	0	−1	4	−1	0	−17
VA.—Lynchburg	4	6	11	7	19	16	10	−6	7	−3	−6
VT.—Northfield	..	−21	−24	−32	−22	−17	−19	−27	−31	−17	−32
WASH. Olympia	23	2	−2	20	7	21	24	28	21	27	−2
WASH. Spokane	14	−11	−30	−10	−23	−10	−5	−19	−2	8	−30
W. VA.—Parkersburg	..	.	12	4	4	8	0	−11	−4	−8	−11
WIS.—La Crosse	−25	−29	−42	−23	−23	−24	−20	−26	−19	−24	−42
WYO.—Cheyenne	−19	−13	−27	−16	2	−7	−29	−4	−17	−20	−29

TABLE LV.

WIND VELOCITY.

Weisbach defines winds as follows:

Scarcely appreciable wind	90 feet per minute equals	1.02	miles per hour
Very feeble wind	180 feet per minute equals	2.04	miles per hour
Feeble wind	360 feet per minute equals	4.1	miles per hour
Brisk wind	1080 feet per minute equals	12.3	miles per hour
Very brisk wind..................	1800 feet per minute equals	20.4	miles per hour
High wind	2700 feet per minute equals	30.7	miles per hour
Very high wind	3600 feet per minute equals	40.1	miles per hour
Violent wind	4200-5400 feet per minute equals	47.8-61.4	miles per hour
Hurricane	6000 feet per minute equals	68.1	miles per hour

The United States Weather Bureau defines a gale as a wind blowing 40 miles per hour.

CHIMNEY FLUES.

For small heating plants the following table, reprinted from Furnace Heating, by the same author as this treatise, may be found useful.

TABLE LVI.

THE APPROXIMATE GRATE SURFACE OR FIRE POT AREA FOR CHIMNEYS OF VARIOUS SIZES AND HIGHTS, BASED ON A RATE OF COMBUSTION OF FIVE POUNDS OF HARD COAL PER SQUARE FOOT OF GRATE SURFACE PER HOUR.

Diameter of chimney. Inches.	Hight of chimney in feet. 40	50	60	70	80	Square or rectangular Flue.
	Approximate grate surface. Square feet.					
8	4	5	6	7	8	8 x 8
10	7	8	9	11	12	8 x 12
12	9	11	13	15	16	12 x 12
14	13	15	17	19	20	12 x 16
16	17	19	21	23	24	16 x 16
18	21	23	25	27	29	16 x 20
20	27	30	33	36	38	20 x 20
22	35	39	43	47	50	20 x 24
24	44	49	54	58	62	24 x 24

TABLE LVII.

AREA OF FIRE POT IN SQUARE FEET.

Diameter. Inches.	Area. Square feet.	Diameter. Inches.	Area. Square feet.
18	= 1.76	28	= 4.27
19	= 1.97	29	= 4.59
20	= 2.18	30	= 4.90
21	= 2.40	31	= 5.25
22	= 2.64	32	= 5.58
23	= 2.88	33	= 5.93
24	= 3.13	34	= 6.30
25	= 3.40	35	= 6.67
26	= 3.68	36	= 7.06
27	= 3.98		

The above table was deduced from formula and table in Babcock & Wilcox's book entitled "Steam." On the basis of 8,000 B. T. U. utilized per pound of coal burned and 250 B. T. U. per square foot of direct steam radiation and 150 B. T. U. per square foot of hot water radiation, the number of square feet grate surface stated in table may readily be converted to direct radiating surface, to which it is adapted by multiplying the square feet grate surface (average firepot area) by 5 x 8,000 and dividing by 250 or 150 for steam or water, respectively.

Mains and branches are to be allowed for as radiating surface. This is expressed as follows:

$$\frac{\text{Grate area} \times 5 \times 8000}{250 \text{ for steam or } 150 \text{ for water}} = \text{Capacity of chimney expressed in square feet of radiating surface.}$$

TESTS ON THE RATE OF COMBUSTION IN FURNACES AND THE VELOCITY OF AIR IN THE PIPES.

The following tests were made on the heating apparatus in a frame house 29 by 35 feet, with parlor, dining room and reception room on the first floor, and four bedrooms and a bathroom on the second floor, heated during one winter season by a brick lined wrought iron furnace with a 22-inch fire pot, and during the following season by a cast iron furnace with a tapering fire pot having an average diameter of about 23 inches.

The brick lined furnace was tested during a 20 days' run in midwinter. The average outside temperature during this period, based on readings taken night and morning, was 26.3 degrees; total weight of coal burned, 2,328 pounds; rate of combustion per square foot of grate per hour, 1.84 pounds. A cold day run was made a little later in the season, the thermometer ranging from 7 degrees below zero to 8 degrees above. During the 24 hours test coal was fed six times, the total weight amounting to 258 pounds, making the average rate of combustion 4.07 pounds per square foot of grate per hour.

The cast iron furnace was tested during a 32 days' trial, the average outside temperature based on three readings per day, being 27½ degrees. The total weight of coal burned was 4,350 pounds; the average per square foot of grate per hour being 1.97

pounds. During this test a record of room temperatures was kept, the average being fully 70 degrees.

A COLD DAY TEST.

During this test a particularly severe day occurred, the temperature falling to 12 below zero. The coal burned during these 24 hours amounted to 300 pounds, giving an average rate of 4.35 pounds per square foot of grate per hour. Coal was fed seven times. The fire pot was red hot while the thermometer remained below zero. The weight of ashes and unconsumed fuel passing through the grate was 10 per cent. of the weight of Lehigh egg coal supplied. The house in which these furnaces were installed was of ordinary frame construction, shingled on building paper and plastered inside. The total cubic contents of rooms connected with the furnace was 11,674 cubic feet. The total combined exposed wall and glass surface was 1,683 square feet.

It is to be noted that both furnaces used were inside the average rating given by reputable manufacturers to furnaces of their size—namely, about 14,000 cubic feet. If based on the exposure such furnaces are expected to carry approximately 1,700 square feet of combined wall and glass surface when the latter does not exceed, say, one-sixth the total exposure. The exposure in this case is practically the same as the above figure. The house had storm windows on the north and west sides, yet an average rate of combustion of nearly 5 pounds per square foot of grate per hour was found necessary to keep the rooms comfortable in severe weather. This high rate requires pretty frequent attention and should be considered a maximum.

DATA ON SIZE OF ROOMS, PIPES, AND THE FLOW OF AIR.

The dimensions and other data of the several rooms are as follows:

TABLE LVIII.

ANEMOMETER TESTS.—FURNACE HEATING.

Rooms. First floor.	Dimensions.—Feet.	Approximate contents. Cubic feet.	Sides exposed.	Size of register.	Diam. of pipe.
Dining room	13 x 18 x 8½	2,000	2	10 x 14	10
Parlor	14½ x 15 x 8½	1,850	2	10 x 14	10
Hall	14 x 18 x 8½	2,140	2	10 x 14	10
Second floor.					
Bedroom	9 x 12 x 8	864	2	8 x 12	7
Bedroom	10 x 19 x 8	1,520	2	8 x 12	8
Bedroom	10 x 12 x 8	960	1	8 x 12	7
Bedroom	13 x 13 x 8	1,350	2	9 x 12	8
Bath	6 x 7½ x 8	390	1	7 x 10	6
		11,674			

Anemometer tests were made with the following results:

Room. First floor.	Temperature at register. Deg. F.	Velocity in pipe. Feet.	Size pipe. Inches.	Horizontal run. Feet.	Elbows. 90°	Elbows. 45°
Dining room	116	418	10	8	1	1
Parlor	114	429	10	2	..	2
Hall	146	465	10	4	1	1
Second floor.						
Bedroom	100	252	7	16	2	2
Bedroom	104	320	8	12	2	2
Bedroom	104	510	7	2	1	1
Bedroom	127	570	8	2	1	1
Bath	103	286	6	8	1	1

The above tests were made with cold air box wide open and with little or no wind. The outside temperature was 5 degrees. The register temperatures were lower than would have been necessary to keep the rooms comfortable had it not been that they had been warmed to a temperature considerably in excess of 70 degrees, and furnace drafts were checked to reduce the heat.

Other tests were made, closing all registers on the first floor, giving velocities of over 500 feet in the rooms on the second floor most remote from the furnace. Tests were made in 34 degree weather, showing a velocity of only about 280 feet in rooms on the first floor. Anemometer readings taken in the cold air box showed a velocity of over 300 feet and a volume of 900 to 980 cubic feet per minute, corresponding to an air change in the rooms heated once in about 13 minutes.

OTHER TESTS.

Tests made in another house with outside temperature 24 degrees showed velocities in pipes leading to the first floor ranging from 306 to 334 feet, the temperature at the registers ranging from 104 to 109 degrees. Pipes leading to the second floor showed velocities in excess of 450 feet per minute with slightly lower register temperatures than on the first floor. The furnace in this case had a 22 inch fire pot. The total volume of air supplied to the house per minute was 850 cubic feet.

Still another test, made in a different house, gave these results for rooms located on the second and third floors, the test being made in cold winter weather. It will be noted that the register temperatures in this case are much higher than in the previous tests:

TABLE LIX.

FLUE VELOCITIES.—FURNACE HEATING.

Room.	Temperature of entering air. Deg. F.	Velocity in pipe. Feet.	Size pipe. Inches.	Horizontal run. Feet.	Elbows.
Parlor	138	250	6 x 10 oval.	9	3
Library	120	210	6 x 7½ oval.	4	2
Dining room	140	275	7 diameter.	15	2
Hall	151	450	6 x 8 oval.	7	2
Bath	108	280	6 diameter.	8	2
Bedroom	152	500	4½ x 7½ oval.	4	3
Rear bedroom	140	540	5 x 7 oval.	12	3

These tests give only a general idea of what velocities may be expected under ordinary working conditions. From the above and other data the writer has adopted these velocities in making furnace heating computations.

Approximate velocity in pipes leading to first floor, 280 feet per minute; to second floor, 400 feet per minute; to third floor, 500 feet per minute.

During the test made in weather 12 degrees below zero the temperature of the air delivered by the furnace was 113 to 115 degrees. When the outside temperature rose to 6 or 8 below zero 122 degrees were indicated by the thermometer placed at register nearest the furnace. The maximum increase in temperature noted was 130 degrees. The wind was blowing strongly into

a wide open cold air box. Had this been partially closed the maximum temperature would doubtless have exceeded 140 degrees, which is commonly used as a basis for computations in work of this kind.

ADVANTAGES OF AIR SUPPLY AT RELATIVELY LOW TEMPERATURES.

There are advantages in supplying air at, say, 120 degrees in zero weather. There is less tendency for the air to remain at the ceiling than when admitted at a higher temperature, thus promoting a better circulation in the room and a nearer approach to a uniform temperature throughout. On the other hand, the lower the temperature of the air supply the greater must be the volume to supply the number of heat units necessary to make good the loss through exposed walls and glass, consequently the more frequent the air change and the greater the fuel consumption.

TABLE LX.

SPACE OCCUPIED BY ANTHRACITE (HARD) COAL PER LONG TON (2240 POUNDS).

Prepared by the author from figures obtained from the Philadelphia and Reading Coal and Iron Company.

Lump	40.6 cu. ft.
Broken	39.4 "
Egg	38.8 "
Stove	38.5 "
Nut	38.4 "
Pea	42.1 "

INDEX.